RESEARCH REPORT - USER'S MANUAL

FOR

COMPUTER PROGRAM AT81Y005

MAY, 1981

Planetsys, A Computer Program For The Steady State and Transient Thermal Analysis of a Planetary Power Transmission System

CONTRIBUTORS:

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SKF REPORT NO. AT81D044

SUBMITTED TO:

NATIONAL AERONAUTICS & SPACE ADMINISTRATION
LEWIS CENTER
2100 BROOKPARK ROAD
CLEVELAND, OH 44135
UNDER CONTRACT NO. NAS3-22690

SUBMITTED BY:

SKF TECHNOLOGY SERVICES
SKF INDUSTRIES, INC.
KING OF PRUSSIA, PA 19406

PROPRIETARY NOTICE

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16. Abstract

The material presented in this manual is structured to guide the user in the practical and correct implementation of PLANETSYS, which is capable of simulating the thermomechanical performance of a multistage planetary power transmission.

In this version of PLANETSYS, the user can select either SKF or NASA models in calculating lubricant film thickness and traction forces

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FOREWORD

The PLANETSYS computer program (20) was originally developed under U.S. Army Contract DAAD05-74-C-0747, sponsored by the Ballistics Research Laboratory (BRL) at Aberdeen Proving Grounds with Mr. W. Thompson as Technical Monitor. The PLANETSYS program simulates the thermo-mechanical performance of a multi-stage planetary power transmission.

This user's manual describes the use of a version of the PLANETSYS computer code developed under NASA-Lewis Research Center Contract, with Mr. H. Coe as Technical Monitor. The revisions made to the program include:

> Implementation of two versions of code within the program - the SKF version and the NASA version. The differences between the two versions reside in the calculation of the elastohydrodynamic (EHD) film thickness and traction forces which develop between rolling element-raceway and rolling element-cage concentrated contacts. The original film thickness model (Dowson-Higginson (13)) and the Tallian traction model (11) are used in the SKF version, while the NASA version uses the Loewenthal model (21) to calculate film thickness and the Allen model (22) to determine traction forces.

Additionally, a new subroutine, FLMFAC (replacing LRHS), is used to determine the lubricant life factor as a function of Λ (film thickness/surface roughness). The values of the lube life factor produced by FLMFAC adhere closely to the curve recommended by the ASME (23, 24).

A new section, Appendix F, describes the differences between the SKF and NASA methods of calculating film thickness and traction forces, and explains the differences in executing each version of the code.

1. INTRODUCTION

In a planetary power transmission system, there are usually three or four planet gears, equally spaced around a sun gear and encircled by a ring gear. In airborne transmissions it is common practice to support the planet gears on single or double row roller bearings such that the planet gear and bearing outer ring comprise a single component. The planet bearings are connected by a planet carrier at the bearing inner rings. Depending on the direction of the power flow, there are six possible kinematic inversions, which the planetary system may adopt. These are described in Table 1.

The major function of the program is to compute for any of the six kinematic inversions, performance characteristics of a planet bearing. The latter may be a cylindrical or spherical roller bearing with an outer ring rigid or flexible. The bearing may contain one or two rows of rollers and its inner ring is taken as an elastic solid.

The Main Program consists of the following major subprograms:

1) Bearing Analysis - Largely based upon methods of Liu and Chiu (1,2), the bearing outer ring is either rigid or flexible. Effects of the latter are reflected in fatigue life calculations and load distributions.

2) Bearing Dimensional Change Analysis - Based on the methods of Timoshenko, (3), and adapted to the bearing system by Crecelius, (4).

3) Generalized Steady State and Transient Temperature Mapping and Heat Dissipation Analyses - based on the methods of Harris, (5), Fernlund, (6), and Andreason, (7).

In this version of the program, the user can select which of the following two models are employed in the calculation of EHD film thickness and concentrated contact traction:

I. SKF Model

1) An elastohydrodynamic (EHD) film thickness model that accounts for i) thermal heating in the contact inlet using a regression fit to results obtained by Cheng (8) and ii) lubricant film starvation using theoretical results derived by Chiu (9).

2) A semi-empirical model for fluid traction in an EHD contact (10), is combined with an asperity load sharing model developed by Tallian (11) to yield a model for traction in concentrated contacts that reflects the state of lubrication as it varies from dry, through partial EHD to the full EHD regime.

II. NASA Model

1) In this empirical model, an elastohydrodynamic (EHD) film thickness is determined from the equation developed by Loewenthal (21). As in the SKF model, film reduction factors due to contact inlet thermal heating and starvation effects are applied.

2) A model for fluid traction in an EHD concentrated contact was developed by Allen (22), based on the fluid shear stress distribution acting over the contact.

Both versions of PLANETSYS also make use of the following bearing related models:

1) Three components of bearing cage related friction are treated for both lubricated and dry operation. For dry operation coulomb friction is assumed. With lubrication, hydrodynamic lubricant shear models are used as follows:

    a) Cage web-roller friction according to methods of Dowson (13)
    b) Cage rail-roller end using the methods for a plain bearing from Marks (14)
    c) Cage rail-ring land using the methods for a short journal bearing according to Dubois and Ocvirk (15).

2) A model for the effect on bearing fatigue life of the ratio of EHD plateau film thickness to composite surface roughness is incorporated (23, 24).

Additionally, models for temperature viscosity and pressure viscosity variation as functions of temperature given by Walther, (19) and Fresco, (12) respectively, were adopted.

The purpose of the program is to calculate the steady state or transient thermal performance characteristics of an airborne type, planetary power transmission system, comprised of up to three stages. Each stage consists of a unique sun, ring and usually three or four planet gears plus a carrier.

To save weight, the typical airborne planetary is designed such that the planet gear and its support bearing form a single, non-separable piece of hardware, as shown in Fig. 1a.

It has been demonstrated in ref. (1 and 2) that the gear tooth loading can affect the bearing rolling element-raceway loading due to the flexibility of the planet gear. This in turn affects the heat generation rate of the planet bearing and thus the thermal performance of the entire system.

The program is structured to handle up to three planetary stages. The power flow through each stage is assumed to be identical. To account for the uniqueness of each stage, a spearate set of input data is required to describe the components of each stage. The total power through the stage is assumed to be shared equally among all planets and all planets are assumed to behave identically.

Each stage may operate according to one of the six kinematic inversions presented in Table 1. Note that the program can handle each inversion and properly accounts for all planet gear and rolling element centrifugal forces which arise from the kinematics.

The planet system thermal performance, loading, frictional heat generation and dimensional stability are all cross-coupled within the program in order to produce a complete picture of system performance.

The types of planet bearings which can be analyzed are shown in Fig. 1b.

TABLE 1

PLANETARY INVERSION INDICES

Inversion No.	Power Input	Fixed	Power Output
1	Sun	Carrier	Ring
2	Ring	Carrier	Sun
3	Sun	Ring	Carrier
4	Carrier	Ring	Sun
5	Ring	Sun	Carrier
6	Carrier	Sun	Ring

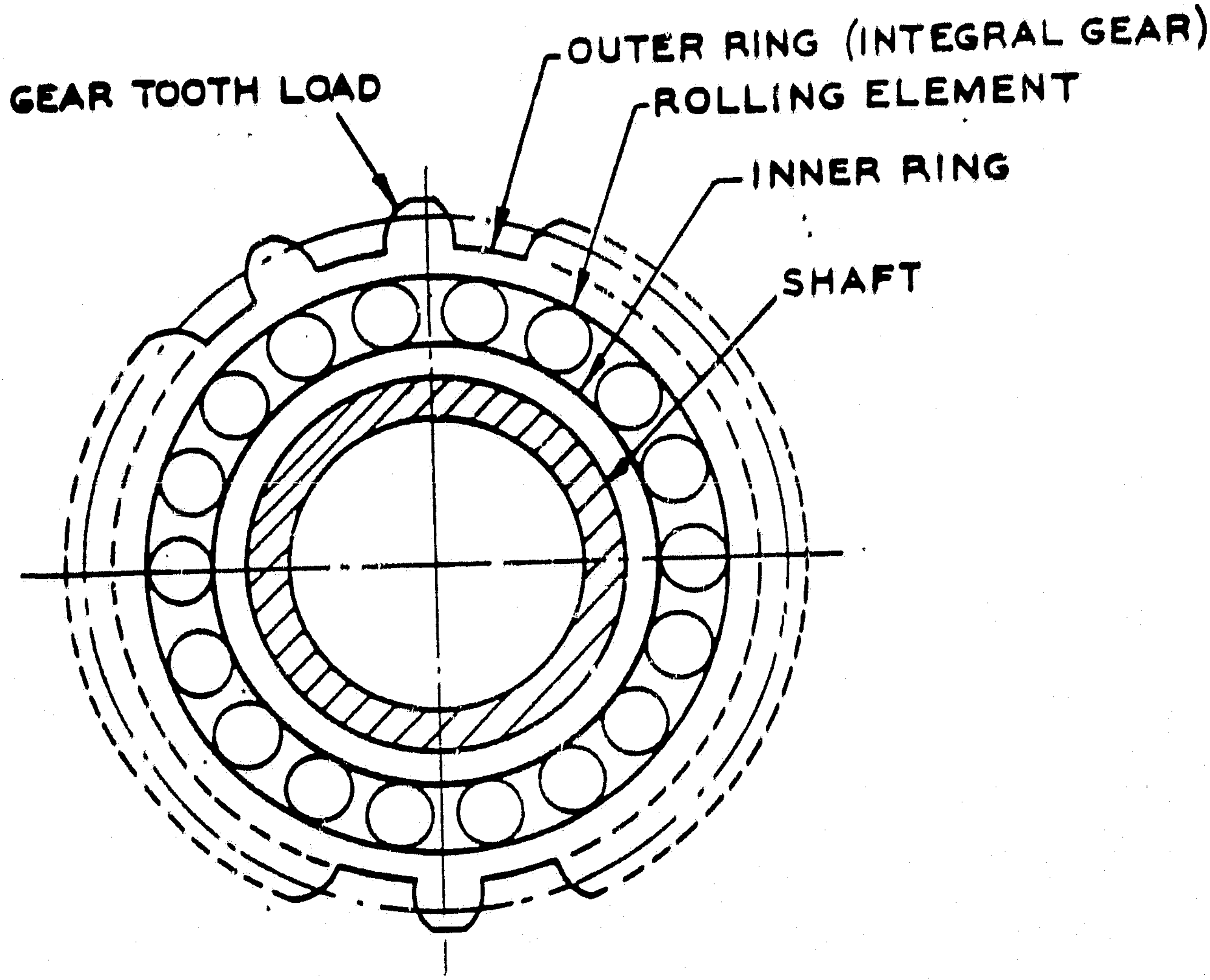

FIGURE 1a PLANET GEAR BEARING SIDE VIEW

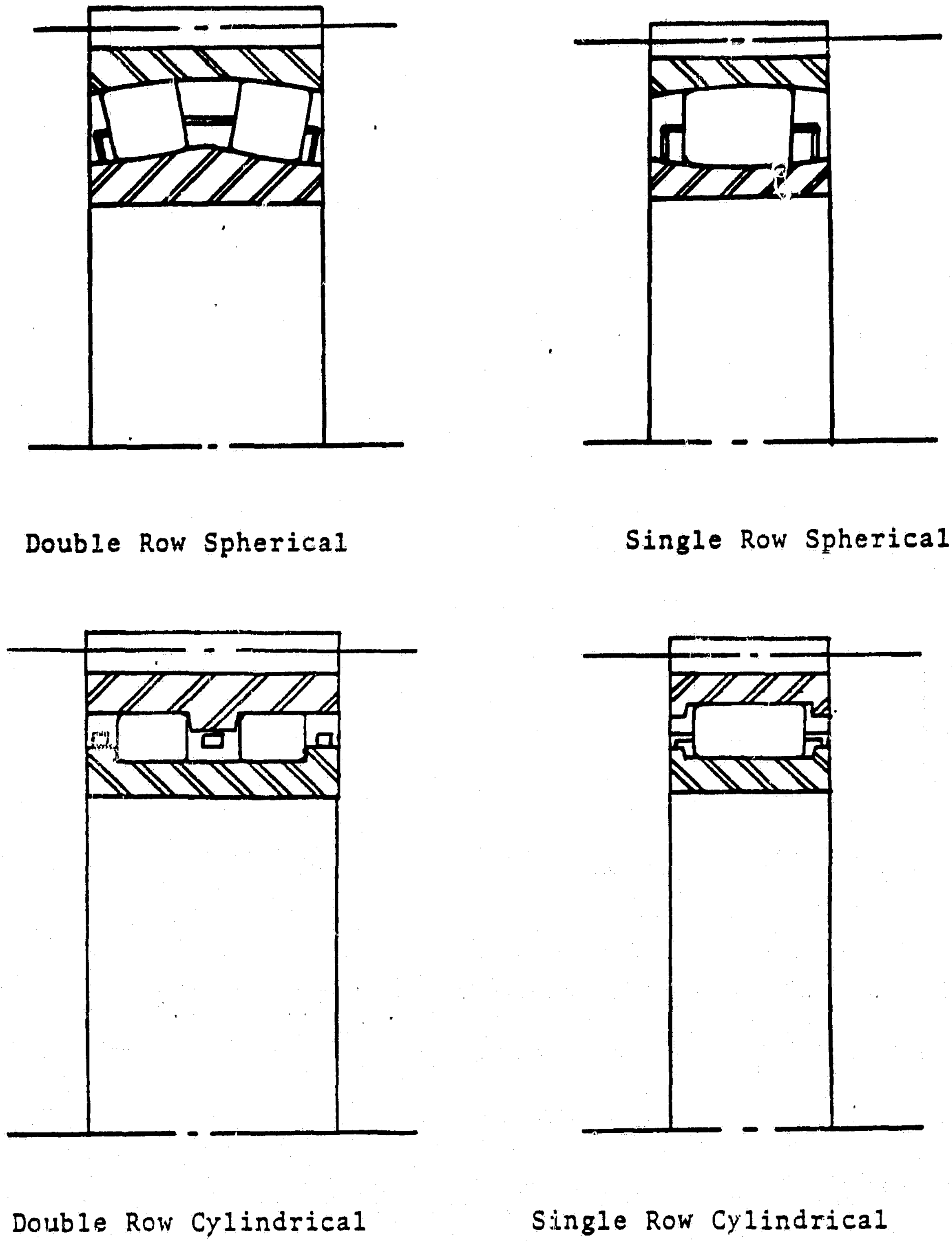

FIGURE 1b PLANET BEARING TYPES WHICH CAN BE ANALYZED WITH "PLANETSYS"

2. PROGRAM INPUT

2.1 Types of Input Data

A complete set of input data comprises data of 3 distinct categories. Within these categories, cards which convey specific kinds of information are referred to as card types. The categories are listed below.

I. Title Cards

A title card plus a second card which provides the program control information for the planet-bearing solution.

II. Bearing Data Cards

A set of up to twenty (20) card types describing one bearing in the assembly. All bearings in the planetary stage are treated identically.

III. Thermal Data Cards

A set of up to nine (9) card types to describe the thermal model of the assembly.

The input data instructions are given in Appendix A, and are for the most part, self explanatory. The units used for input data are also given in Appendix A. They are presented in an eighty column data format. A description of the variables is given in the input instruction forms.

2.2 Data Set I - Title Cards

2.2.1 Title Card

This card should contain the computer run title and any information which might prove useful for future identification. The full eighty (80) columns are available for this purpose. The title will appear at the top of each page of Program output.

2.2.2 Flag Set Card

ITEM 1

Number of reduction stages, can be 1 thru 3.

ITEM 2

Print Flag (NPRINT), equal to zero is normal and will result in no intermediate or debug output. With a value of one, a low level intermediate print is obtained at the end of each planet-bearing iteration. The values of the variables, the inner ring displacements and the equation residues are printed.

At the end of each bearing iteration, wherein the rolling element and cage equilibrium equations are solved, an error parameter is printed which has the value:

$$\text{Error Parameter} = \Delta X_N / X_{N-1}$$

ΔX_N is the change in the variable X specified at iteration N.

X_{N-1} is the value of the variable specified at the previous iteration.

The Error Parameter is calculated for each of the bearing variables, but only the largest one is printed.

Additionally, at the end of each Clearance Change iteration, the clearance change error parameter is printed. This error is defined:

$$\text{Error Parameter} = \frac{DCL_N - DCL_{N-1}}{\text{Rolling Element Diameter}}$$

Where DCL_N and DCL_{N-1} denote the clearance changes calculated at the current and previous iterations respectively.

If NPRINT is set at 2 all the above information is printed. Additionally, the variable values and residue values are printed for each iteration of the rolling element equilibrium solution.

ITEM 3

Fit calculation flag (ITFIT), ITFIT controls the number of iterations allowed to satisfy the bearing clearance change iteration scheme. If ITFIT is set to zero (0) or left blank, the clearance change portion of the program is not executed. If a positive number is input, the clearance change scheme is utilized with a maximum iteration limit of five (5). If a negative number is input, the scheme is used with a maximum iteration limit equal to the absolute value of the negative number.

ITEM 4

Main Loop iteration flag (ITMAIN), ITMAIN limits the number of iterations attempted during the solution of the bearing equilibrium problem, i.e. establishing the equilibrium of bearing reactions and applied gear tooth and inertial loads. If ITMAIN is set to a positive integer, then (20) iterations are permitted. If ITMAIN is set to a negative number, the number of iterations is limited to the absolute value of that number. If ITMAIN is set to zero or left blank, only one iteration is performed.

ITEM 5

Fit Loop Accuracy (EPSFIT), EPSFIT is the convergence criterion for the diametral clearance change portion of the analysis. As mentioned under item 2 above, this error parameter is defined to be:

$$\text{Error Parameter} = \frac{DCL_N - DCL_{N-1}}{\text{Rolling Element Diameter}}$$

The iteration scheme is terminated when the error parameter is less than the input value of EPSFIT. If EPSFIT is left blank or is set to zero (0), the program default value of 0.0001 is used. In the above expression, DCL is the change in bearing clearance, and N refers to the iteration number.

ITEM 6

If Plot Flag is set at either "T" or ".True." a plot of radial deflection and load vs. azimuth angle will be produced at the end of the program output. Default is false (No plot).

ITEM 7

Units Flag (IMET), IMET permits input data to be in either International or English units. A value of 1 indicates International, 0 or blank English units. Dimensions of specific variables are given in Appendix A.

ITEM 8

Material property Signal (IMT), a value of 1 allows input of material constants. If IMT is 0 or blank, the program assumes the material properties of steel apply. The default values are listed in sections 2.3.14 thru 2.3.17.

2.3 Data Set II - Bearing Data

Most of the input instructions are self-explanatory. Where more explanation than given in the input data format instructions is required items are treated on an individual basis by card type and item number.

2.3.1 Data Card No. 3 - Rolling Element Information

ITEM 1

Bearing type, columns 1-10 must be specified, left justified, i.e., "S" or "C" in column 1. This format must be followed since the Program recognition of bearing type, (Spherical or Cylindrical rolling bearing), is derived from reading the "S" or "C" in the first column of this card.

ITEM 8

Roller Skew Angle, columns 71-80, is used to study the effect of roller skew on bearing generated heat and is not used in calculation of bearing equilibrium. Input is in degrees.

2.3.2 Data Card No. 4 - Bearing Data

ITEM 4

Refers to the number of slices into which the roller raceway contact may be divided. A maximum value of twenty, (20) is permitted. A default value of two (2) is used if Item 4 is blank or zero. Each slice is the same width.

ITEM 5

Columns 71 thru 80 contain a signal, termed the crown drop flag, which specifies whether the roller-race crown drops will be calculated, or read directly. If item 5 is blank or zero, the crown drops are calculated based on the roller-race crown radius, and flat length input information. If the crown drop flag is other than zero or blank the nonuniform separation of the roller and raceway must be specified at the center of each slice. The slice widths are identical. The non-uniform roller-raceway separation is input on data cards 7 and 7A. (see Fig. 2)

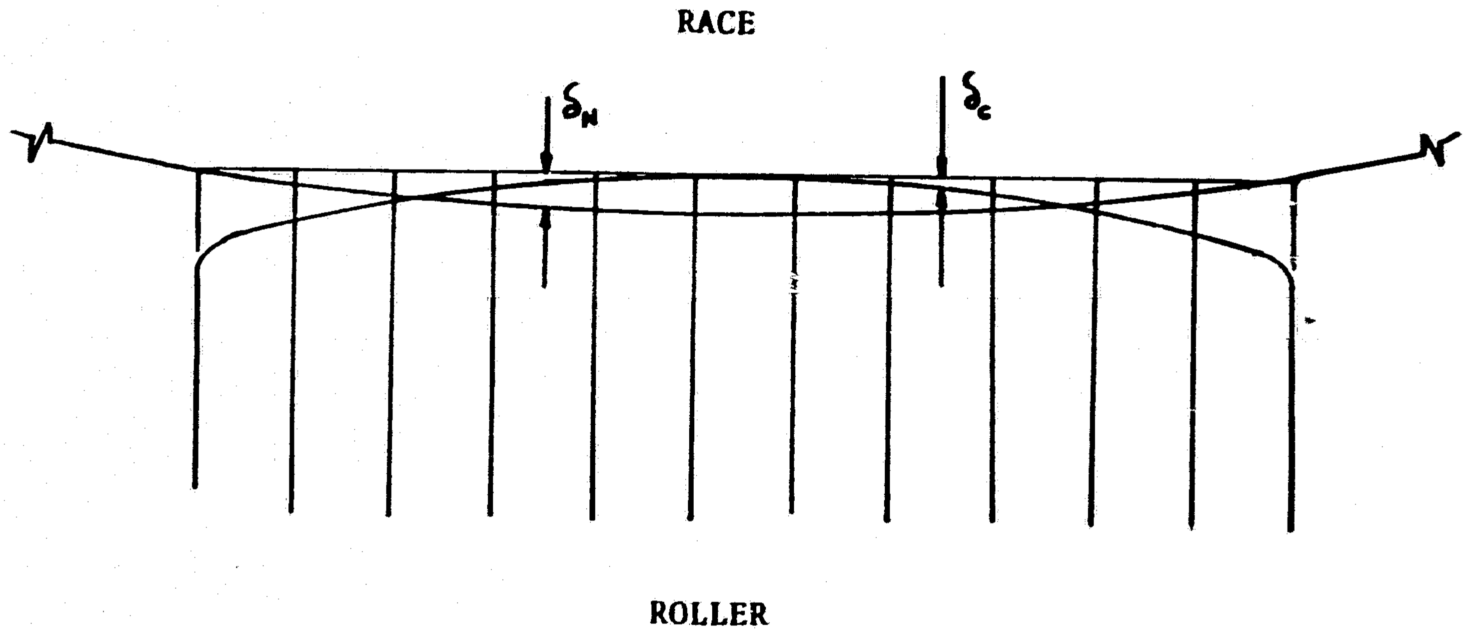

FIGURE 2 ROLLER-RACE LAMINATION SHOWING RELATIVE APPROACH (δ n) AND CROWN DROP (δ c)

2.3.3 Data Card 5 - Inner Raceway Information

ITEM 1

Columns 1-20, "Steel Designation" for the inner ring. Alphameric-literal description of the steel type such as "M-50" or "AISI 52100" is input.

ITEM 2

"Effective Contact Length" refers to the longest possible length which can be obtained at a roller-race contact. Typically this is the roller total length less the corner radii. If, however, the raceway undercuts are exceptionally large so that the track width is smaller than the effective roller length then the track width should be input.

ITEM 5

Columns 51 thru 60, the number input for item 5 is used to account for improved materials, multiplying the raceway fatigue lives as determined by Lundberg-Palmgren methods. Typical life factor values for modern steels are in the neighborhood of 2.0 to 6.0. In the ASME Publication Life Adjustment Factors for Ball and Roller Bearings, the Material Factor D and the Material Process Factor E should be used multiplicatively as inputs for item 5. The program computes a lubricant life factor based on the value of Λ (EHD plateau film thickness divided by RMS composite surface roughness). The lube life factor is calculated by sub-routine FLMFAC using a least squares fit to the curve recommended by ASME (23, 24). This factor ranges in value from 0.21 for $\Lambda \leq 0.6$ to 3.0 for $\Lambda \geq 10.0$.

2.3.4 Data Card 5A - Outer Raceway Information

Data Card 5A, is the same as 5, but outer raceway information is given.

2.3.5 Data Card 6 - Surface Data

Items 1 through 6 define the statistical surface microgeometry parameters of the rollers and raceways. Items 1 through 3 require the input of center line average (CLA) surface roughness. Within the program CLA values are converted to RMS by multiplying by 1.25.

Items 4 through 6 are RMS values of the slopes, (degrees) of the surface asperities. These are measured in a traverse across the groove for rings, longituidinally for rollers. Typical values for raceway and rolling element surfaces are 1 to 2 degrees.

2.3.6 Data Cards 7 & 7A - Crown Drops

These cards are used to input the separation between the inner and outer raceways and the roller at the center of each slice along the roller profile with the high points of the roller and race in contact, i.e. with all clearance between roller and raceway removed. These cards must be omitted if item 5 of Data Card No. 4 is 0 or blank.

2.3.7 Data Card 8 - Cage Information

ITEM 7

Assumed cage slip, input as a positive number less than 1, is used in the calculation of roller-inner raceway rolling element speeds. As the cage slips, sliding velocity at the contact is increased. Typical slip values range from 0.0 to 0.15.

2.3.8 Data Card No. 9 - Diametral Clearance Iteration Data

ITEM 1

Number of Clearance Iterations, although not used in the calculations, is an input item. It should be set at 1., and is included only for future parametric studies of system performance as a function of bearing diametral clearance.

ITEM 2

Change in clearance for next iteration, set at 0., is not included in any calculations, (see ITEM 1).

2.3.9 Data Card No. 10 - Planet Gear Data

ITEM 2

Planetary Inversion Index, refers to the specific kinematic condition under which the system operates. Valid numbers are one (1) thru six (6), the inversions are as follows:

Inversion No.	Power Input	Fixed	Power Output
1	Sun	Carrier	Ring
2	Ring	Carrier	Sun
3	Sun	Ring	Carrier
4	Carrier	Ring	Sun
5	Ring	Sun	Carrier
6	Carrier	Sun	Ring

Note: For planetary inversions 3-6, where the planet carrier is not fixed, the inertial load of the planet gear is assumed to be totally taken by the planet carrier so that the gear tooth load remains unaffected.

ITEM 6

Moment of inertia of outer ring cross section (I) is calculated for input relative to the centroidal axis (Rc) as shown in Fig. 3. The shaded area in Fig. 3 indicates the effective area used in the calculation of I. In most cases the neutral axis and centroidal axis are coincident and standard equations which relate I to cross section geometry can be used. The user may specify that the outer ring be rigid, (with respect to radial deflection) by inputting the moment of inertia greater than 10^5 IN^4 (4.2×10^6 CM^4).

2.3.10 Data Card No. 11 - Misc. Information

ITEM 1

Power thru stage refers to the total power through the stage. Internally, the program will divide this number by the total number of planet gears in the stage before calculating the gear tooth loads.

2.3.11 Data Card No. 12 - Shaft Fit Data

This card is to be included only if the change in bearing diametral clearance with operating conditions is to be calculated, i.e., if item 3 (ITFIT) on Data Card No. 2 is non-zero. On input data, tight interference fits bear a positive sign and loose fits a negative sign.

Item 3 on Card No. 12 is termed the shaft effective width. The value specified for the effective width may be as large as twice the ring width.

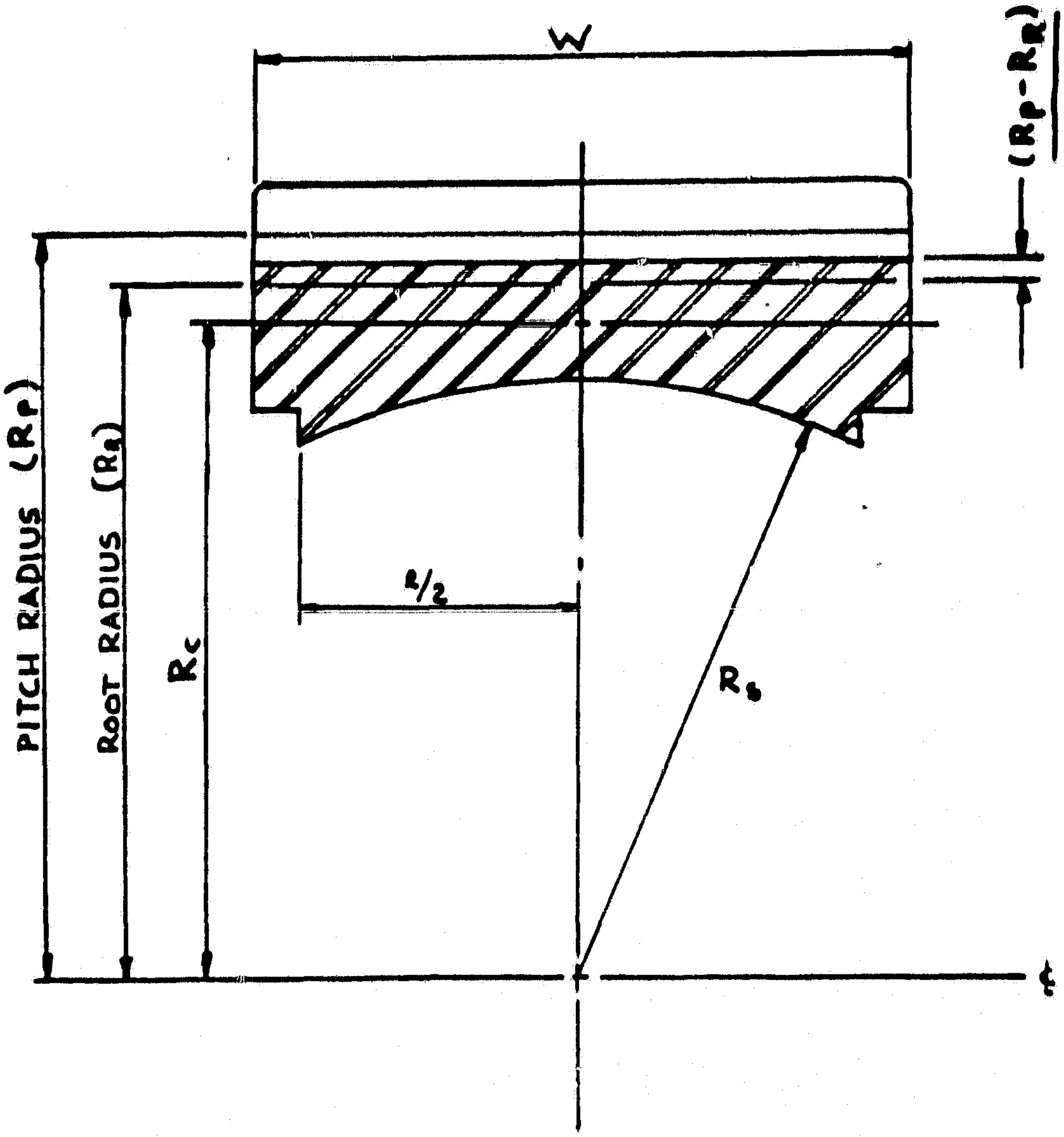

FIGURE 3 PLANET GEAR OUTER RING CROSS SECTION

Use of an effective width is an attempt to account for the greater radial rigidity of a shaft onto which a ring is pressed. This is due to the fact that the shaft deflects over a distance that extends beyond the ring width. In the program the calculated internal pressure on the ring due to its interference fit with the shaft, is distributed over the shaft effective width. Using double the actual width as the effective width is customary.

2.3.12 Data Card No. 13 - Shaft Fit Dimensions

These items are self explanatory.

2.3.13 Data Card No. 14 - Material Properties

This card defines the modulus of elasticity for the shaft, inner ring, rolling element and planet gear respectively. This card is to be included only if the material properties signal on Title Card No. 2 is 1. A default value of 204083 N/mm^2 (29.6x10^6 PSI) is used.

2.3.14 Data Card No. 15 - Material Properties

This card defines the Poisson's ratio for the shaft, inner ring, rolling element and planet gear, respectively. This card is to be included only if the material properties signal on Title Card No. 2 is 1. A default value of 0.30 is used.

2.3.15 Data Card No. 16 - Material Properties

This card defines the density for the shaft, inner ring, rolling element and planet gear respectively. This card is to be included only if the material properties signal on Title Card No. 2 is 1. A default value of 7.806 g/cm^3 (0.282 lb/in^3) is used.

2.3.16 Data Card No. 17 - Material Properties

This card defines the coefficient of thermal expansion for the shaft, inner ring, rolling element and planet gear respectively. This card is to be included only if the material properties signal on Title Card No. 2 is 1. A default value of 12.24×10^{-6} 1/°C (6.8×10^{-6} 1/°F) is used.

2.3.17 Data Card No. 18 - Lubrication and Friction Data

ITEMS 1 & 2

Items 1 and 2 are the amounts by which the combined thickness of the lubricant film on the rolling track and rolling element is increased during the time interval between the passage of successive rolling elements, from whatever replenishment mechanisms are operative. Item 1 applies to the outer and Item 2 to the inner race-rolling contacts respectively. If Item 1 is zero or blank the mode of friction is assumed to be dry.

At the present time the magnitude of the inner and outer raceway replenishment layers has not been correlated to lubricant flow rate, lubricant application methods and bearing size and speed factors. The user is required to establish proper values for the replenishment layer thickness. As a rough guide the following suggestions are made:

1) To avoid starvation, the replenishment layer thicknesses should be one to two times the EHD film thickness which develops in the rolling element raceway contacts.

2) Because of centrifugal force effects, intuition suggests that the outer raceway replenishment layer should be several times thicker than that prescribed at the inner raceway.

ITEM 3

Item 3, XCAV, describes the percentage of the bearing cavity estimated by the user to be occupied by the lubricant. $0 \leq XCAV \leq 100$. This term is used in an equation which estimates the amount of frictional heat generated by lubricant churning.

As with the replenishment layer thicknesses, the amount of free lubricant should be correlated with lubricant flow rate, lubricant application methods and bearing size and speed factors. At this time such correlations do not exist. XCAV values less than five percent are recommended.

ITEMS 4 & 5

Items 4 and 5 are the coefficients of coulomb friction applicable for the contact of asperities. If Items 1 and 2 are zero, signifying dry operation, then items 4 and 5 serve as the coulomb friction coefficients which prevail at respective contacts.

2.3.18 Data Card No. 19 - Lubricant Type

This card is omitted if Item 1 on card 18 is zero or blank which implies dry friction.

The relevant lubricant data for four specific lubricants (see Table 2) has been coded into PLANETSYS. The lubricant input information consists only of a single number which designates the particular lubricant but the relevant information for the lubricant is printed in the input data list.

TABLE 2

PROPERTIES OF FOUR LUBRICANTS

LUBRICANT NO.	LUBRICANT TYPE	KINEMATIC VISCOSITY (cs) 37.78° C (100°F)	KINEMATIC VISCOSITY (cs) 98.89°C (210°F)	DENSITY @15.56°C (60°F) gm/cm^3	THERMAL CONDUCTIVITY W/m/°C	THERMAL COEFF. OF EXPANSION 1.°C 10^{-4}
1	Mineral Oil	64.0	8.0	0.88	0.116	6.336
2	MIL-L-7808G	17.8	3.2	0.95	0.152	7.092
3	Polyphenal Ether	25.4	4.13	1.20	0.119	7.470
4	MIL-L-23699	28.0	5.1	1.01	0.152	7.452

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If Item 1, NCODE is 1, 2, 3, or 4 PLANETSYS uses preprogrammed lubricant properties and no further information is required.

NCODE	Lubricant
1	SHELL TURBO 33
2	MIL-L-7808G
3	Polyphenyl-Ether
4	MIL-L-23699

NCODE may also be specified as negative (-1 to -4), in which case the traction characteristics of the respective lubricant NCODE noted above are used but the actual properties specified by the Items on data cards 20 & 21 override the hard coded data. This option is most useful in specifying various mineral oils i.e. NCODE = -1.

If the NASA traction model is being used, two additional lubricant data items should be specified on card no. 21:

a) AKN = Empirical Lubricant Constant - col. 11-20
b) FRIC = Lubricant friction coefficient - col. 21-30

Default values of AKN = 50. and FRIC = .07 will be used if the items are left blank.

For lubricants NCODE = 1 through 4 the following values of AKN and FRIC are assigned:

NCODE	AKN	FRIC
1	18.2	.075
2	18.2	.045
3	24.9	.070
4	18.2	.070

2.4 Data Set III - Thermal Model Data

Subsequent to each calculation of bearing generated heat rates, either the steady state or transient temperature mapping scheme may be executed. What follows is a description of input variables. A more complete description of the thermal portion of "PLANETSYS", along with some heat transfer information is given in Appendix E.

2.4.1 Data Card No. T1 - System Control Information

Data Card T1 is a control card. If no temperature map is to be calculated, this card is to be included as a blank card followed by Data Card T2. Data Card T1 contains control input for both steady state and transient thermal analyses. It is not intended, however, that both analyses be executed with the same run.

ITEM 1

The highest node number (M). The highest node number must not exceed one hundred, (100).

ITEM 2

Number of unknown temperature nodes (N). This number should equal the total number of unknown temperatures. It is required that all nodes with unknown temperatures be assigned the lowest consecutive numbers starting from one (1). The program assumes that all node numbers greater than N (from N+1 to M) represent known boundary temperatures.

ITEM 3

Common Initial Temperature (TEMP)^{o}C: The temperature solution iteration scheme requires a starting point, i.e., guesses of the equilibrium temperatures. Data Card T3 allows the user to input guesses of individual node temperatures. When a node is not given a specific initial temperature, the temperature specified as Item 3 of Data Card T1 is assigned.

ITEM 4

Punch Flag (IPUNCH): If the Punch Flag is not zero (0) or blank, the system equilibrium temperatures along with the respective node numbers will be punched according to the format of Data Card T3. This option is useful if the user makes a steady state run with lubrication and then wishes to use the resultant temperatures for initialization of a transient dry friction run. The latter is performed to assess the consequence of lubricant flow termination.

ITEM 5

"Output Flag" (IUB). If the "Output Flag" is not zero the bearing program output and a temperature map will be printed after each call to the planet bearing solution scheme.

This printout will allow the user to observe the flow of the solution and to note the interactive effects of system temperatures and bearing heat generation rates. Since the temperature solution is not mathematically coupled to the bearing solution the possibility exists that the solution may diverge or oscillate. In such a case, study of the intermediate output produced by the "Output Flag" option may provide the user with better initial temperature guesses that will effect a steady state solution.

ITEM 6

"Maximum Number of Calls to the planet bearing Program" (IT1). IT1 is the limit on the number of Thermal-planet-bearing iterations. The user must input a non-zero integer such as 5 or 10 for the Program to iterate to an equilibrium condition. If IT1 is left blank or set to zero (0) or 1, planet bearing performance will be based on the initially guessed temperatures of the system. The temperatures printed out will be based on the bearing generated heats. It is unlikely that an acceptable equilibrium condition

will be achieved. However, the temperatures which result may provide better initial guesses for a subsequent run than those specified by the user.

IT1 also serves as a limit on the transient temperature solution scheme, by limiting the number of times the planet bearing solution scheme is called. Each call to the planet bearing scheme will input a new set of bearing heats to the transient temperature scheme until a steady state condition is approached or until the allowed transient solution time limit is reached.

ITEM 7

"Absolute Accuracy of Temperatures for the External Solution" (EP1). In the steady state thermal solution scheme, each calculation of system temperatures occurs after a call to the planet-bearing routine which produces bearing generated heats. After the system temperatures have been calculated for each iteration, using the temperature mapping heat dissipation scheme, each node temperature is checked against the nodal temperature at the previous iteration.

If $\{t_{n,i} - t_{n-1,i}\} \leq EP1$ for all nodes i, then equilibrium has been achieved and the iteration process stops. Where $t_{n,i}$ = temperature of ith node at nth thermal iteration; $t_{n-1,i}$ = temperature of ith node at n-1th thermal iteration.

ITEM 8

"Iteration Limit for the temperature mapping heat dissipation Solution" (IT2). After each call to the planet bearing program, the node to node heat transfer scheme is solved to determine the steady state equilibrium temperatures throughout the system, based on the calculated set of bearing heat generation rates. If IT2 is left blank or set to zero, the number of iterations permitted to solve the heat transfer scheme is limited to twenty, (20).

ITEM 9

"Accuracy for Internal Thermal Solution" (EP2). The use of EP2 is explained in Appendix E (page E11). If EP2 is left blank or set to zero (0), a default value of 0.001 is used.

ITEM 10

"Starting Time" (START) is a time, T_s, at which the transient solution begins; usually set to zero (0).

ITEM 11

"Stopping Time" (STOP) is the time in seconds at which the transient solution terminates, T_f. The transient solution will generate a history of the system performance which will encompass a total elapsed time of

$(T_f - T_s)$ seconds.

ITEM 12

"Calculation Time Step" (STEPIN). The transient internal solution scheme solves the system of equations:

$$t_{k+1} = t_k + Q_k \Delta t/\rho CpV$$

$$\Delta t = \text{STEPIN}$$

The user may specify STEPIN. If left blank or set to zero (0), the Program calculates an appropriate value for STEPIN using the procedure described in Appendix E (pp. E11 - E13).

ITEM 13

"Time Interval Between Printed Temperature Maps" (TTIME) seconds. The user must specify the length of time which will elapse between each printing of the temperature map. The interval will always be at least as large as the "calculation timestep" (STEPIN).

ITEM 14

"Time Interval Between Calls of the Planet Bearing Portion of the Program" (BTIME). BTIME will always have a value larger than or equal to (STEPIN) even if the user inadvertently inputs a shorter interval. Computational time savings result if BTIME is greater than STEPIN; however, accuracy might be lost.

2.4.2 Data Card T2 - Bearing Temperatures

Card Type T2 is required, one card for each planetary stage, if no thermal analysis is being performed. The temperature data is used within the planet-bearing analysis to fix temperature dependent properties of the lubricant.

For this purpose the inner race, outer race and lubricant bulk cavity temperatures are used in the analysis which calculates the change in bearing diametral clearance from "off the shelf" to operating conditions. Temperatures must be given in °C.

2.4.3 Data Card T3 - Nodal Temperatures

In the steady state analysis this card is used to input initial guesses of individual nodal temperatures for unknown nodes as well as the constant temperatures for known nodes, such as ambient air and/or an oil sump.

In the transient analysis, Data Card T3 is used to input the nodal temperatures of all nodes at (START) = T_S, i.e., at the initiation of the transient solution.

2.4.4 Data Card T4 - Bearing Node Numbers

With this card, node numbers are assigned to the components of each bearing, one card per stage. With this information the proper system temperatures are carried into each respective bearing component to be used in

the bearing analysis. The inner race and inner ring node numbers may or may not be the same at the user's discretion. Similarly the outer race and outer ring node numbers may or may not be the same.

2.4.5 Data Card T5 - Heat Generating Nodes

Card type T5 is required, one card for each planetary stage, if a thermal analysis is to be performed. The planet bearing system analysis accounts for frictional heat generated at five locations in the bearing, i.e. at the inner race, the outer race, between the cage rail and ring land, at the cage rolling element contacts and in the bulk lubricant due to drag. This card allows the heat generated to be distributed equally to two nodes. For instance, the heat generated at the inner race-rolling element contact should be distributed half to the rolling element and half to the inner race. The heat developed between the cage and the inner ring land may be distributed half to the inner ring and half to the cage, if a cage node has been defined, otherwise, half to the bulk lubricant.

2.4.6 Data Card T6 - Nodes with Constant Power Source

This card specifies the node numbers and the heat generation rate for those nodes where heat is generated at a constant rate such as at rubbing seals.

The heat generated by the planet-sun and planet-ring gear contacts is not currently calculated by PLANETSYS. In order to do a thermal analysis of a planetary bearing assembly, the heat generation rates for these contacts must be calculated by the user and input using T6 data card (S). The heat distributed to the planetary gear node should correspond to a single planet contact. The heat distributed to the non-planetary nodes (sun gear, ring gear) should correspond to the total for all planet contacts.

2.4.7 Data Card T7 - Heat Transfer Coefficients

This card type is used to input the numerical values of the various heat transfer coefficients which appear in the equations for heat transfer by conductivity, free convection, forced convection, radiation and fluid flow. Up to ten coefficients of each type may be used. Separate values of each type of coefficient

are assigned an index number via card T7 and in describing heat flow paths (Data Card T8 below) it is necessary only to list the index number by which heat transfers between node pairs.

Indices 1-10 are reserved for conduction, 11-20 for free convection, 21-30 for forced convection, 31-40 for radiation and 41-50 for fluid flow (product of specific heat, density and volume flow rate).

As an example, for heat transfer by conduction within steel having a thermal conductivity of 53.7 watts/m°C, one could prepare a card type T7 with the digit 1 punched in column 10 and the value 53.7 punched in the field corresponding to card columns 11-20. If a conduction coefficient of 46.7 were applicable for certain other nodes in the system one could punch an additional card assigning index No. 2 to the value 46.7 by punching a "2" in card column 10 and 46.7 anywhere within card columns 11-20.

Rather than inputting constant forced convection coefficients, optionally, these coefficients can be calculated by the program in one of three ways. If the calculated option is exercised, a pair of cards is used in place of a single card containing a fixed value of α. The contents of the pair of cards depends upon which of the three optional methods are used.

Option 1) α is independent of temperature, calculated as a function of the Nusselet number which in turn is a function of the Reynolds number Re, the Prandtl number Pr as follows, (cf. 17).

$$\alpha = Nu\ \lambda_{oil}/L$$
$$Nu = a Re^b Pr^c$$

Where λ_{oil} is the lubricant conductivity, L is a characteristic length (with a unit of meters) and a, b, and c are constants.

Option 2) α is a function only of fluid dynamic viscosity, and viscosity is temperature dependent.

$\alpha = c\eta^d$
where c and d are constants

Option 3) **α is a function of the Nusselt, Reynolds and Prandtl numbers and viscosity is temperature dependent.**

2.4.8 Data Card T8 - Heat Flow Paths

This card defines the heat flow paths between pairs of nodes. Every node must be connected to at least one other node, i.e., two or more independent node systems may not be solved with a single Program execution.

The calculation of heat transfer areas is based on lengths, L_1 and L_2 input using Card Type 8. Additionally, the type of surface for which the area is being calculated is indicated by the sign assigned to the heat transfer coefficient index. If the surface is cylindrical or circular the index should be positive, if the surface is rectangular the index should be input as a negative integer. Note that due to the axial symmetry of the planetary system most nodes will represent circular rings and thus the heat transfer indices are predominantly positive. For conduction the length L_3 gives the separation distance between the two nodes.

In the case of radiation between concentric axially symmetric bodies, L_3 is the radius of the larger body. For radiation between two parallel flat surfaces or for conduction between nodes, L_3 is the distance between them.

Fluid flow heat transfer accounts for the energy which the fluid transports across a node boundary. Along a fluid node at which convection is taking place, the temperature varies. The nodal temperature which is output is the average of the fluid temperature at the output and input boundaries. If the emerging temperature of the fluid is of interest, it is necessary to have a fluid node at the fluid outlet. At this auxiliary node only fluid flow heat transfer occurs and the fluid temperature would be constant throughout the node. Thus the true fluid outlet temperature will be obtained.

Conduction of heat through a bearing is controlled by index 51. The actual heat transfer coefficients which contain a conductivity, area and a path length term are calculated in the bearing portion of the program. The term is based upon an average outer race and inner race rolling element contact.

Special guidelines must be followed in specifying heat transfer linkages between planetary nodes and non-planetary nodes (e.g., conduction between planet gear node and ring gear node). For conduction, convection, or radiation heat transfer, the planetary node must be defined as node I (columns 11-15). The number of planets must be input in columns 51-60. The heat transfer area defined by L_1 (columns 21-30) and L_2 (columns 31-40) must correspond to a single planet.

For fluid flow heat transfer between a planetary node and a non-planetary node, the data on card T8 depends on the direction of flow. The program assumes that the direction of fluid flow is from node I (columns 11-15) to node J (columns 16-20). When node I is the planet node, the first fluid flow index (columns 1-10) must correspond to the flow from an individual planet. The second flow index (columns 21-30) must correspond to the total flow from all planets. The number of planets is specified as a <u>positive</u> number in columns 51-60. When node J is the planet node, the first fluid flow index must correspond to the total flow to all planets. The second index must correspond to the flow to an individual planet. The number of planets is specified as a <u>negative</u> number in columns 51-60.

2.4.9 Data Card T9 - Node Heat Capacity

This card inputs data required to calculate the heat capacity of each node in the system. This card type is required only for a transient analysis.

3. COMPUTER PROGRAM OUTPUT

3.1 Introduction

The Program Output is intended to provide the user with a complete picture of the planet bearing system performance.

This portion of the Manual is not meant to be a list and explanation of all program output. Most items are self explanatory and have been omitted from further discussion.

Two sample program outputs are included in Appendices B and C, reflecting use of the NASA and SKF traction models, respectively. In both cases, the output represents the solution for a single stage planetary gear system transmitting 202.2 HP.

The first five pages of output essentially consist of a summary of the input data categorized into bearing, cage, steel, lubricant and loading data. The remaining six pages contain the calculated program output.

3.2 Bearing Output

3.2.1 Gear Tooth and Inertial Loads

Figure 4 shows a single gear tooth and the applied loads. The gear tooth torque arises as a result of the tangential tooth load with a lever arm equal to the difference in radii between the planet pitch circle and planet ring neutral axis.

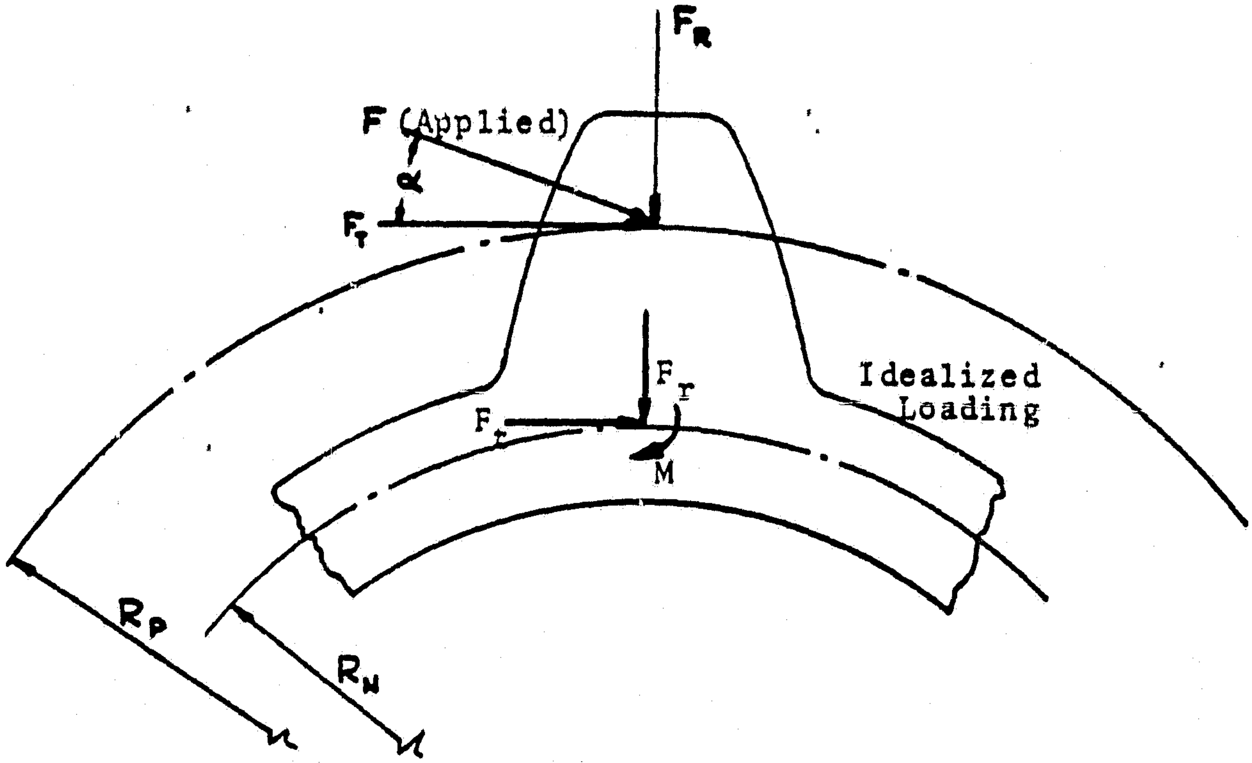

FIGURE 4 IDEALIZED GEAR TOOTH LOADS

3.2.2 Fatigue Lives

The L_{10} fatigue life of the outer and inner raceways as well as the bearing are calculated based upon the 4th power relationship.

The bearing life represents the statistical combination of the two raceway lives and in the case of a double row bearing, the combination of the two single row lives which are assumed to be equal. These lives reflect the combined effects of the lubricant film thickness and material life factors.

3.2.3 h/sigma

The ratio h/σ also referred to as Λ , is printed for the most heavily loaded rolling element. The variable h, represents the EHD plateau film thickness with thermal and starvation effects considered. The variable σ, represents the composite root mean square surface roughness of the rolling element and the relevant raceway.

3.2.4 Life Multpliers

Lubrication - This life multiplier is a function of h/σ at each concentrated contact. Its value ranges from 0.21 for $\Lambda \leq 0.6$ to 3.0 for $\Lambda \geq 10.0$. This subject is covered in more detail in Section 2.3.3, Item 5.

Material - This output simply reflects the input value.

Again, it is covered in Section 2.3.3.

3.2.5 Temperatures Relevant to Bearing Performance

These temperatures fully describe the temperature conditions which affect the performance of a given bearing. If one of the temperature mapping options is used, the temperatures printed reflect the results of the particular option. If neither temperature option is used, the list is simply a repeat of the input data. Note that there are separate temperatures for outer and inner raceway and ring temperatures. The raceway temperature is used to determine lubricant properties. The ring temperatures are used in the bearing dimension change analysis. The raceway and ring temperatures may be the same value.

3.2.6 Frictional Heat Generation Rate and Bearing Friction Torque

Frictional Heat Generation Rate

The various sources of frictional heat generated within the bearing are listed. The values printed for "OUTER RACE, INNER RACE, R.E. DRAG, R.E. CAGE and CAGE-LAND" represent the sum of the generated heats for all rolling elements and cage. Additionally, the heats printed for the outer and inner raceways plus the rolling element cage, reflect the friction developed outside the concentrated contacts, i.e. the HD friction, as well as the EHD friction developed within the concentrated contacts. "R.E. DRAG" should be interpreted as the heat resulting from lubricant churning as the rolling elements plow through the air-oil mixture. Cage-land heat generation rate is an estimate since unavoidable variations in cage-land diametral clearance have a marked effect on cage-land temperature, making precise calculations impractical. (See ref. 15).

Torque

The torque value is calculated as a function of the total generated heat and the sum of the outer and inner ring rotational speeds. The intent is to present a realistic value of the torque required to drive the bearing.

3.2.7 EHD Film and Heat Transfer Data

EHD Film Thickness

These values refer to the calculated EHD plateau film thickness at both contacts of the most heavily loaded rolling element and include the effects of the thermal and starvation reduction factors.

Starvation Reduction Factor

These factors give for the inner and outer ring contacts, the reduction in EHD film thickness due to lubricant film starvation according to the methods of Chiu, (9).

These factors pertain to the EHD film thickness for both the inner and outer race contacts of the most heavily loaded rolling elements, but are applied to the respective inner and outer race film thickness for each rolling element in the bearing.

Thermal Reduction Factor

These factors are calculated according to the methods of Cheng, (8) and pertain to the EHD film thickness for both the inner and outer race contacts of the most heavily loaded rolling elements, but are applied to the respective inner and outer race film thickness for each rolling element in the bearing.

Meniscus Distance

This parameter is defined as the distance along the direction of rolling from the contact center to the oil meniscus in the contact inlet. It is calculated according to the methods of Chiu [9] for the inner and outer race contacts of the most heavily loaded rolling element.

Raceway-Rolling Element Conductivity

These data reflect the amount of heat transfer between rolling element and raceway per unit degree difference between the two components. These data reflect the average of all outer and inner contacts respectively.

3.2.8 Fit and Dimensional Change Data

Fit Pressures

These data refer to the pressures built up as a consequence of interference fits between shaft and inner ring and housing and outer ring. Pressures are presented both for the standard cold-static condition (16° C) and the operating condition.

Clearances

"Original" refers to cold unmounted clearance which is specified at input if the diametral clearance change analysis is executed. "Change" refers to the change in diametral clearance at operating conditions relative to the cold unmounted condition. A minus sign indicates a decrease in clearance. "Operating" refers to the clearance at operating conditions.

3.2.9 Lubricant Temperatures and Physical Properties

The lubricant properties, particularly the dynamic viscosity and to a lesser degree the pressure viscosity coefficient are heavily temperature dependent. These factors enter the EHD film thickness calculation and the HD and EHD friction models. The lubricant is assumed to be at the same temperature as the relevant raceway. As noted elsewhere, these temperatures may be either input directly or calculated by the Program.

The physical properties printed are self explanatory. The units are enumerated.

3.2.10 Cage Data

Cage-Land Interface

The cage data indicates the performance parameters at the interface between the cage rail and the ring land on which the cage is guided. The torque and heat rate require no explanation. The eccentricity ratio defines the degree to which the cage approaches the ring on which it is guided at the point of nearest approach. The radial displacement of the cage relative to the bearing axis is divided by one half the cage-land diametrical clearance. An eccentricity ratio of one indicates cage-land contact. A ratio of zero indicates that the cage rotation is concentric with the bearing axis. The cage-land separating force results from the rolling element cage loads driving the cage to an eccentric position and causing pressure to develop between the cage rail and ring land. Only the cage-land and rolling element pocket forces are considered in determining the cage eccentricity. The centrifugal force which results from the eccentricity, although available, is not considered in the analysis.

3.3 Rolling Element Output

3.3.1 Rolling Element Orientation

The first rolling element may be located in either of the following ways:

1. When the planet bearing center is stationary as in cases 1 and 2 of Table 1, the first rolling element is encountered at 9 o'clock, gear tooth loads being applied at 12 and 6 o'clock.

2. When the planet bearing center orbits as in cases 3 thru 6 of Table 1, the first rolling element is located at 12 o'clock plus half the rolling element spacing. Again, gear tooth loads are applied at 12 and 6 o'clock.

3.3.2 Rolling Element Raceway Loading

Normal Forces

The gear tooth loads, the inertial load of the planet gear, and the rolling-element contact loads acting on the planet bearing outer ring form a self-equilibrating loading system.

The rolling-element contact loads depend on the bearing outer ring deflections, the rolling-element inertial loads, and the radial displacement of the bearing inner ring, which is considered as an elastic solid.

The cage force is always directed tangent to the bearing pitch circle. If the rolling element orbital speed is positive, a positive cage force indicates that the cage is pushing the rolling element, tending to accelerate it. Cage force is a function of the tangential component of the rolling element inertia less the raceway friction forces.

Hertz Stress

The stress printed represents the maximum normal stress at the most heavily loaded slice of the roller raceway contact.

Outer Ring Deflections

For a flexible outer ring, the radial deflections are given at the various rolling element locations. The sign convention of radial displacement is shown in Figure 5.

3.3.3 Deflection Plotting

A plot showing the outer ring deflections and rolling element - outer race contact loads versus azimuth angle is provided at the User's Option. (See sect. 2.2.2, Item 6)

3.4 Thermal Output

As is the case for bearing output, all of the input data is printed. The calculated output data is presented in the form of a temperature map in which a node number and the respective nodal temperature appear. The appearance of the steady state and transient temperature maps are identical. The transient temperature map also includes the time (T) at which the temperature calculations were made.

4. GUIDES AND LIMITATIONS TO PROGRAM USE

"PLANETSYS" is intended to be as general as possible with the following limits on system size:

1. Number of rolling elements per bearing row must not exceed 40.

2. Gear tooth loads must act at the two ends of the diameter on a line passing through the sun and planet gear centers and in the radial plane containing the bearing center.

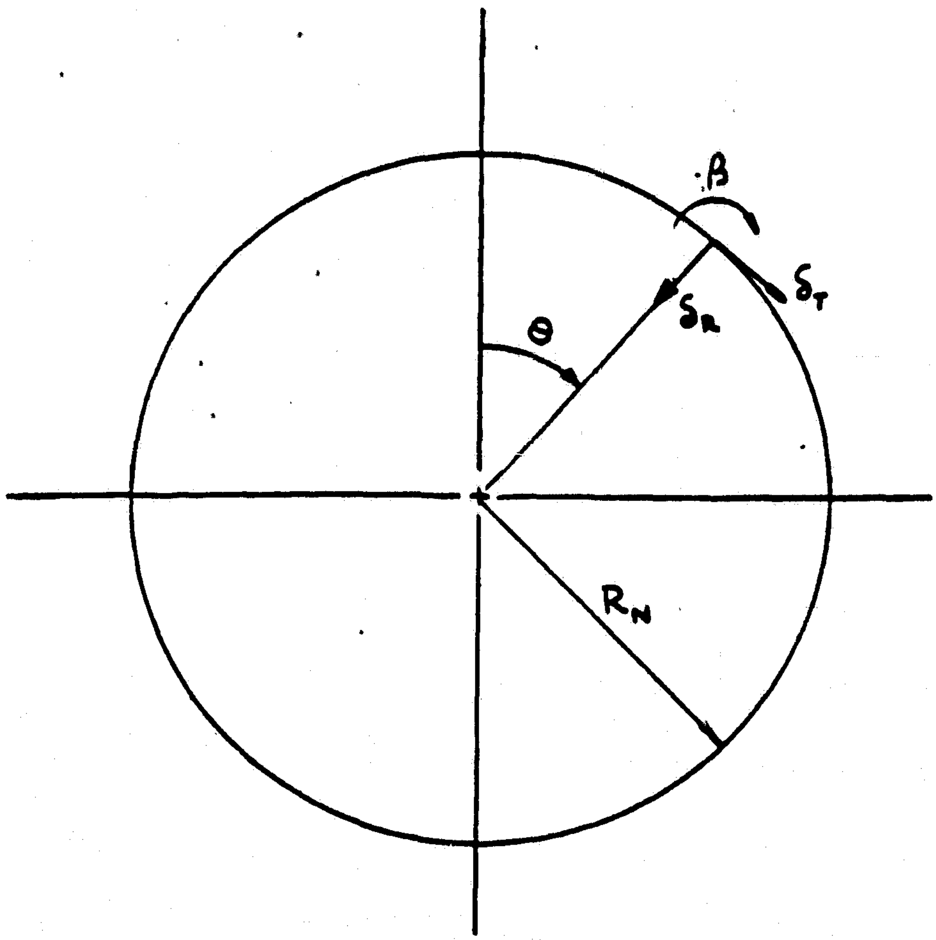

FIGURE 5 POSITIVE DIRECTION OF OUTER RING RADIAL DISPLACEMENT

3. Number of stages in the transmission system is limited to 3.

4. Number of temperature nodes used to describe the system - 100 maximum.

Some general guides for efficient use of the Program are listed below:

1. Attempt to input bearing operating diametral clearance rather than calculate it. Or, execute the diametral clearance change analysis once for a group of similar runs and use the output from the first run as input to the subsequent runs omitting the clearance change analysis.

2. Attempt to input accurate operating temperatures rather than calculate them.

3. The more non-linear the problem the more computer time. In the thermal solutions, if possible, eliminate nonlinearities by omitting radiation terms and by using constant rather than temperature dependent free and forced convection coefficients.

4. In the transient thermal solution, space the calls to the planet-bearing solution (BTIME) to as large an interval as prudently possible. Be careful, however, too long an interval will produce large errors in heat rate predictions.

5. In the steady state thermal analysis attempt to estimate nodal temperatures on a node by node basis. Nodes which are heat sources should have higher temperatures than the surrounding nodes.

It is also suggested that a constant user of the program should study the hierarchical Program flow chart Appendix D, along with the Program listing to gain an appreciation of the program complexity and the flow of the problem solution. The Program is comprised of many small functional subroutines. Knowledge of these small elements may allow the user to more easily piece together the philosophy of the total problem solution.

5.0 LIST OF REFERENCES

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APPENDIX A

"PLANETSYS" INPUT DATA FORMS

DATA CARD # 1

1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25	26	27	28	29	30	31	32	33	34	35	36	37	38	39	40	41	42	43	44	45	46	47	48	49	50	51	52	53	54	55	56	57	58	59	60	61	62	63	64	65	66	67	68	69	70	71	72	73	74	75	76	77	78	79	80
2	0	A	4																																																																												

TITLE TO BE PRINTED ON EACH PAGE

DATA CARD #2 – FLAG SET

Columns	Variable	Format	Description
1–10		F10.0	No. of Reduction Stages – 3 Max.
11–20	NPRINT	F10.0	Debug Flag IF 0 or Blank No Debug Output IF = 1 Main Loop Debug Only IF = 2 Full Debug Output
21–30	ITFIT	F10.0	Fit Calc Flag 0 or Blank No Fit calcs – and no iterations Any positive No. Fit calcs perf. with a max of 5 iterations Neg. No. Fit calcs are performed, No. of iterations equals ABS. value of given number
31–40	ITMAIN	F10.0	Main Loop Iteration Flag Any positive no. Pre-Set Max (20) is used Neg. No. No. of iterations equals ABS. value of given number 0 or Blank One iteration
41–50	EPSFIT	F10.0	Fit Loop Accuracy* 0 or Blank Pre-Set accuracy of .0001 is used
51–60		L10	Plot Flag Blank or "False" No Plots "True" Output will contain plots of Outer Ring deflection vs. azimuth angle in cartesian coordinates
61–70	IMET	F10.0	Dimensions Flag 0 or Blank Input variables are given in English Units 1 Input variables given in SI Units
71–80	IMT	F10.0	Material Properties Signal 1. User will specify mat. properties 0 or Blank Pre-set material properties are used omit data cards 14 through 17 incl.

* Fit Loop Accuracy defined as change in diametrical clearance divided by Rolling Element Diameter.

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AT81D044

ITA CARD #3 - Rolling Element Information

BD (1)	BD (2)	BD (3)	BD (4)	BD (5)	BD (6)	BD (7)	BD (8)
1 2 3 4 5 6 7 8 9 10	11 12 13 14 15 16 17 18 19 20	21 22 23 24 25 26 27 28 29 30	31 32 33 34 35 36 37 38 39 40	41 42 43 44 45 46 47 48 49 50	51 52 53 54 55 56 57 58 59 60	61 62 63 64 65 66 67 68 69 70	71 72 73 74 75 76 77 78 79 80
A 1	F 1 0 . 0	F 1 0 . 0	F 1 0 . 0	F 1 0 . 0	F 1 0 . 0	F 1 0 . 0	F 1 0 . 0
Rolling Element Type	No. of Rows	No. of Rolling Elements per Row (40 Max)	Rolling Element Diameter	Rolling Element Total Length	Roller Profile Radius	Roller End Sphere Radius	Roller Skew Angle Default =0
C in Col 1 - Cylindrical S in Col 1 - Spherical Blank - Cylindrical			Inches / MM	Inches / MM	Inches / MM	Inches / MM	Degrees

A-4

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AT81D044

DATA CARD # 4 - Bearing Data

BB (9)	BB (10)	BB (11)	BB (18)				
F10.0	F10.0	F10.0	F10.0				F10.0
Bearing Contact Angle	Diametral Clearence	Pitch Diameter	No. of Raceway Slices				Crown Drop Read Signal *
Degrees	Inches/ MM	Inches / MM	Maximum of 20 Default Is 2				

* Normally Zero or Blank, if Set at 1, Non-Uniform Roller Raceway geometry is input on cards 7 & 7A.

AT81D044

DATA CARD #5 - Inner Raceway Information

Columns	1–20	21–30	31–40	41–50	51–60	61–70	71–80
Variable		BD(13)	BD(15)	BD(17)	BD(20)		
Format	5A4	F10.0	F10.0	F10.0	F10.0		
Description	Steel Designation - Inner Raceway	Inner Raceway Roller Effective Length	Spherical - Inner Race. Osculation Cylindrical - Rolling Element, Flat Length	Spherical - Leave Blank Cylindrical - Inner Raceway Crown Radius	Inner Raceway Life Adjustment Factor		
Units		Inches/MM	Inches/MM	Inches/MM			

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AT81D044

DATA CARD #5A – Outer Raceway Information

Columns 1–20	Columns 21–30	Columns 31–40	Columns 41–50	Columns 51–60	Columns 61–70	Columns 71–80
	BD(12)	BD(14)	BD(16)	BD(19)		
5 A 4	F 1 0 . 0	F 1 0 . 0	F 1 0 . 0	F 1 0 . 0		
Steel Design for Outer Raceway	Outer Raceway Roller Effective Length	Spherical – Outer Race. Osculation Cylindrical – Leave Blank	Spherical – Leave Blank Cylindrical – Outer Raceway Crown Radius	Outer Raceway Life Adjustment Factor		
	Inches/MM		Inches/MM			

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CARD #6 - SURFACE DATA

	BD (21)	BD (22)	BD (23)	BD (24)	BD (25)	BD (26)		
Columns	1–10	11–20	21–30	31–40	41–50	51–60	61–70	71–80
Format	F10.0	F10.0	F10.0	F10.0	F10.0	F10.0		
Field	Surface Roughness Outer Race	Surface Roughness Inner Race	Surface Roughness Rolling Elt.	Asperity Slope Outer Ring	Asperity Slope Inner Ring	Asperity Slope Rolling Elt.		
Units	Micro Inches/ Microns	Micro Inches/ Microns	Micro Inches/ Microns	Degrees	Degrees	Degrees		

DATA CARD #7 - Outer Raceway Crown Drops   Input only if Crown Drop Flag (Columns 71-80 of Data Card #4) is 1.

1-10	11-20	21-30	31-40	41-50	51-60	61-70	71-80
F10.0	F10.0	F10.0	F10.0	F10.0	F10.0	F10.0	F10.0
Outer Raceway Crown Drop at Laminum 1	Outer Raceway Crown Drop at Laminum 2	... and so on	Use more than one card if Laminae is greater than 8.				

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DATA CARD #7A - Inner Raceway Crown Drops

Input only if Crown Drop Flag (Columns 71-80 of Data Card #4) is 1.

1-10	11-20	21-30	31-40	41-50	51-60	61-70	71-80
F10.0	F10.0	F10.0	F10.0	F10.0	F10.0	F10.0	F10.0
Inner Raceway Crown Drop at Laminum 1	Inner Raceway Crown Drop at Laminum 2	... and so on, Use more than one card if Laminae is greater than 8.					

DATA CARD #8 - Cage Information

BD(44)	BD(45)	BD(46)	BD(47)	BD(48)	BD(49)	BD(27)	
1–10	11–20	21–30	31–40	41–50	51–60	61–70	71–80
F10.0	F10.0	F10.0	F10.0	F10.0	F10.0	F10.0	
Cage Type	Rail-Land Diameter	Rail Width	Ball Land Dia. Clearance	Cage Pocket Clearance	Cage Weight	Assumed Cage Slip Default -0	
-1. Outer Ring Land Riding 0. Rolling Element Riding +1. Inner Ring Land Riding	Inches/MM	Inches/MM	Inches/MM	Inches/MM	Lbs/Kilos	Fraction	

DATA CARD #9 - Bearing Diametral Clearance Iteration Data

HD(28)	HD(29)
Columns 1–10	Columns 11–20
F 1 0 . 0	F 1 0 . 0
Number of Clearance Iterations	Change in Clearance for Next Iteration

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DATA CARD #10 -

BD(32)	BD(33)	BD(38)	BD(39)	BD(40)	BD(41)	BD(42)	BD(43)
1–10	11–20	21–30	31–40	41–50	51–60	61–70	71–80
F10.0	F10.0	F10.0	F10.0	F10.0	F10.0	F10.0	F10.0
No. of Planetary Bearings in Stage	Planetary Invers. Index	Pitch Dia. of Planet Gear	Width of Planet Gear	Dia. of Planet Neutral Axis	Planet Gear Moment of Inertia	Gear Tooth Pressure Angle	Planet Outer Ring Weight
	See Table 1	Inches/MM	Inches/MM	Inches/MM	IN^4/CM^4	Degrees	Lbs/Kg

*If set greater than $10^5 IN^4$, Ring Assumed to be Rigid

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DATA CARD #11 - Speeds & Misc. Information

BD(31)	BD(34)	BD(35)	BD(36)	BD(37)
Columns 1–10	Columns 11–20	Columns 21–30	Columns 31–40	Columns 41–50
F10.0	F10.0	F10.0	F10.0	F10.0
Power Thru Stage	Avg. Speed of Sun	Avg. Speed of Carrier	Avg. Speed of Ring	Pitch Dia. of Sun
HP/KW	RPM	RPM	RPM	Inches/MM

DATA CARD #12 - FIT Data   Include only if IIFIT, Data Card #2, is nonzero.

BD(51)	BD(52)	BD(53)	BD(54)	BD(55)	
Columns 1–10	Columns 11–20	Columns 21–30	Columns 31–40	Columns 41–50	Columns 51–80
F10.0	F10.0	F10.0	F10.0	F10.0	
Shaft Fit Positive If Interference	Housing Fit Positive if Interference	Shaft Effective Length	Bearing Inner Ring Width	Width of Planet Gear	
Inches/MM	Inches/MM	Inches/MM	Inches/MM	Inches/MM	

DATA CARD # 13 - FIT Data   Include only if LTFIT, data card #2, is nonzero.

BD(57)	BD(58)	BD(59)	BD(60)	BD(61)	BD(62)		
1 2 3 4 5 6 7 8 9 10	11 12 13 14 15 16 17 18 19 20	21 22 23 24 25 26 27 28 29 30	31 32 33 34 35 36 37 38 39 40	41 42 43 44 45 46 47 48 49 50	51 52 53 54 55 56 57 58 59 60	61 62 63 64 65 66 67 68 69 70	71 72 73 74 75 76 77 78 79 80
F 1 0 . 0	F 1 0 . 0	F 1 0 . 0	F 1 0 . 0	F 1 0 . 0	F 1 0 . 0		
Shaft Inner Diameter	Shaft O.D	Bearing Bore Dia.	Inner Ring Mean O.D.	Planet Bearing Avg. I.D.	Planet Bearing Avg. O.D.		
Inches/MM	Inches/MM	Inches/MM	Inches/MM	Inches/MM	Inches/MM		

DATA CARD #14 - Material Properties - Modulus of Elasticity   Include only if IHT, data card #2, is 1.

BD(66)	BD(67)	BD(68)	BD(69)
F10.0	F10.0	F10.0	F10.0
MODULUS OF ELASTICITY			
Shaft	Inner Ring	Rolling Element	Planet
PSI/N per MM^2	PSI/N per MM^2	PSI/N per MM^2	PSI/N per MM^2

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DATA CARD # 15 - Material Properties - Poisson's Ratio   Include only if IHT, data card #2, is 1.

BD(71)	BD(72)	BD(73)	BD(74)
F10.0	F10.0	F10.0	F10.0
Poisson's Ratio			
Shaft	Inner Ring	Rolling Element	Planet

DATA CARD #16 - Material Properties, - Density   Input only if INT, data card #2, is 1.

Columns 1–10	Columns 11–20	Columns 21–30	Columns 31–40	Columns 41–80
BD(76)	BD(77)	BD(78)	BD(79)	
F10.0	F10.0	F10.0	F10.0	
Density				
Shaft	Inner Ring	Rolling Element	Planet	
$Lb/in^3/gm/cm^3$	$Lb/in^3/gm/cm^3$	$Lb/in^3/gm/cm^3$	$Lb/in^3/gm/cm^3$	

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DATA CARD # 17- Material Properties - Coefficient of Thermal Expansion
Input only if INT, data card #2, is 1.

BB(01) (cols. 1–10)	BB(02) (cols. 11–20)	BB(03) (cols. 21–30)	BB(04) (cols. 31–40)
F10.0	F10.0	F10.0	F10.0

Coefficient of Thermal Expansion			
Shaft	Inner Ring	Rolling Elt.	Planet
1/F°/1/C°	1/F°/1/C°	1/F°/1/C°	1/F°/1/C°

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DATA CARD #10- Lube Data

Columns	1–10	11–20	21–30	31–40	41–50	51–60	61–70	71–80
Variable	BD(102)	BD(103)	BD(104)	BD(106)	BD(107)			
Format	F10.0	F10.0	F10.0	F10.0	F10.0			
Description	Lubricant Replenishment Layer Thickness – Outer Raceway	Lubricant Replenishment Layer Thickness – Inner Raceway	Percent Lubricant In Bearing Cavity	Raceway-Rolling Element Asperity Friction Coefficient	Cage Frict. Coefficient			
Units	Inches/MM	Inches/MM						

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DATA CARD #19 - Lube Code Omit this card if the outer raceway replenishment layer thickness data card #18 is 0 or blank.

BD(93)

1	2	3	4	5	6–80
F	1	0	.	0	

Lube Type
(See Table 1)

-4.
-3.
-2.
-1.
0.
} Include Next 2 Cards

1.
2.
3.
4.
} Omit Next 2 Cards

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DATA CARD #20 - Lube Properties

Omit this card if the outer raceway replenishment layer thickness, data card #18, is 0 or blank. Also omit this card if lube type, data card #19, is 1-4.

	BD(91)	BD(92)	BD(94)	BD(95)	BD(96)	BD(97)
9A2	F10.0	F10.0	F10.0	F10.0	F10.0	F10.0
Lube Designation	Low Temperature	High Temperature	Kinematic Viscosity at "Low Temp"	Kinematic Viscosity at "High Temp"	Density of Lube at 15.5°C	Coefficient of Thermal Expansion
	°F/°C	°F/°C	IN^2 per Sec/ Centistoke	IN^2 per Sec/ Centistoke	Lb per In^3 / GM per CM^3	$°F^{-1}$/ $°C^{-1}$

DATA CARD #21 - Lube Properties

Omit this card if outer raceway replenishment layer thickness, data card #18, is 0 or blank. Also omit the card if lube type data card #19, is 1-4. Read this data if NCODE $\leq$ 0.

BD(98)

Columns 1–10	Columns 11–20	Columns 21–30	Columns 31–80
F10.0	F10.0	F10.0	
THERMAL CONDUCTIVITY BTU/FT – F^{o}-HR WATTS/M-C^{o}	AKN EHD HIGH CONTACT-STRESS FACTOR	FRIC ALLEN FRICTION COEFFICIENT	

INPUT THESE VALUES WHEN EXECUTING NASA VERSION (SEE PAGE 23 AND APPENDIX F)

DATA CARD NO. 11 - SYSTEM CONTROL INFORMATION

THE CONTROLLING CARD. IF NO TEMP. CALCULATION IS DESIRED, THEN LEAVE THE CARD BLANK.

Variable	N	M	TEMP	IPUNCH	IOS	IT1	EP1	IT2	EP2	START	STOP	STEP IN	TINT	BINT
Format	I5	I5	F5.0	I5	I5	I5	F5.0	I5	F5.0	F5.0	F5.0	F5.0	F5.0	F5.0
Group	GENERAL						STEADY STATE ONLY			TRANSIENT ONLY				
Description	HIGHEST NODE NUMBER	HIGHEST UPSTREAM NODE NUMBER, SAME AS NUMBER OF UPSTREAM NODES	COMMON INITIAL TEMPERATURE. SELECTED TEMPERATURES CAN BE GIVEN DIFFERENT INITIAL VALUES USING CARD 13. (°C)	PUNCH FLAG, USUALLY ZERO. IF ≠ 0 FINAL TEMPERATURES WILL BE PUNCHED ACCORDING TO THE FORMAT OF CARD 13. THEN, THEY CAN BE READ IN AS INITIAL TEMPERATURES IN A LATER RUN.	OUTPUT FLAG, USUALLY ZERO. IF ≠ 0 BEARING OUTPUT AND A TEMPERATURE MAP WILL BE PRINTED EVERY TIME THE SHAFT-BEARING PROGRAM HAS BEEN CALLED.	MAXIMUM NO. OF CALLS OF THE planet BEARING PROGRAM. For steady state analysis the max. no. of thermal iterations to be performed. For transient analysis, a value of N* given by equ. below should be input.	ABSOLUTE ACCURACY OF TEMPERATURES (°C)	ITERATION LIMIT, USUALLY LEFT BLANK. IF ≠ 0 THE INTERMEDIATE ITERATION LIMIT. IF BLANK PRE-SET LIMIT IS USED	ACCURACY USUALLY LEFT BLANK. IF ≠ 0 IT IS THE INTERMEDIATE ALL. ACCURACY	STARTING TIME (SEC)	STOPPING TIME (SEC)	CALCULATION TIME STEP. IF LEFT BLANK, THE PROGRAM WILL CALCULATE A SUITABLE STEP. (SEC)	TIME INTERVAL BETWEEN PRINTED TEMP. MAPS. THE INTERVAL WILL ALWAYS BE AT LEAST EQUAL TO THE CALC. TIME STEP. (SEC)	TIME INTERVAL BETWEEN CALLS OF SHAFT-BEARING PROGRAM. ALWAYS AT LEAST EQUAL TO THE CALC. TIME STEP. (SEC)

$$* \; N = IFIX\left(\frac{Stop - Start}{B\ Time}\right) + 1$$

Where IFIX indicates that digits to the right of decimal point are truncated.

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DATA CARD T2 – BEARING TEMPERATURES One card is required for each Planetary Stage. Temperatures must be input in °C. Use only if no temperature calc. is desired, and then give no more thermal data.

Columns	1–5	6–10	11–15	16–20	21–25	26–30	31–35	36–40	41–45	46–50	51–55	56–60	61–65	66–70	71–80
Format	F5.0	F5.0	F5.0	F5.0	F5.0	F5.0	F5.0	F5.0	F5.0	F5.0	F5.0	F5.0			
Field	Shaft	Inner Ring	Inner Race	Rolling Elt.	Outer Race	Planet Ring	Cage	Bulk Lube							

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DATA CARD T3 - NODAL TEMPERATURES
Thermal Data - Individual Initial Temperatures (^{o}C)
Use As many cards as needed, followed by a blank card

Node Number	Initial Temperature	Etc.													Node Number	Initial Temperature
I5	F5.0	I5	F5.0	I5	F5.0	I5	F5.0	I5	F5.0	I5	F5.0	I5	F5.0		I5	F5.0

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DATA CARD T4 – BEARING NODE NUMBERS   Include one T4 card for each planetary stage.

I5	I5	I5	I5	I5	I5	I5	I5
Shaft	Inner Ring	Inner Race	Rolling Element	Outer Race	Planet Ring	Cage	Bulk Lube

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DATA CARD T5 - HEAT GENERATION NODES

Include one T5 card for each planetary stage.

Columns 1–5	Columns 6–10	Columns 11–15	Columns 16–20	Columns 21–25	Columns 26–30	Columns 31–35	Columns 36–40	Columns 41–45	Columns 46–50	Columns 51–55	Columns 56–60	Columns 61–65	Columns 66–70	Columns 71–75	Columns 76–80
I5	I5	I5	I5	I5	I5	I5	I5	I5	I5						
Inner Race		Outer Race		Cage-Ring Land		Roller Drag		Roller-Cage							
Half the heat is generated at this node	Half the heat is generated at this node														
The above two node numbers can be the same number.															

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DATA CARD T6

Thermal Data - Nodes with a constant power source

USE as many cards as needed, followed by a blank card

1–5	6–10	11–15	16–20	21–25	26–30	31–35	36–40	41–45	46–50	51–55	56–60	61–65	66–70	71–75	76–80
I5	F5.0	I5	F5.0	I5	F5.0	I5	F5.0	I5	F5.0	I5	F5.0	I5	F5.0	I5	F5.0
Node Number	Power (Watts)	Node Number	Power (Watts)	....and	so forth	....	....	....	....	....	....	....	....	Node Number	Power (Watts)

DATA CARD T7

HEAT TRANSFER COEFFICIENTS Input one or two cards per coefficient. Use as many T7 cards as required <u>followed by a blank card</u>.

Columns 1–10	11–20	21–30	31–40	41–50	51–60	61–70	71–80
I10	F10.0	F10.0	F10.0	F10.0	F10.0	F10.0	
INDEX FOR LATER IDENTIFICATION OF HEAT TRANSFER COEFF. 1-10 (Conduction) 11-20 (Free Conv.) 21-30 (Forced Conv*) 31-40 (Radiation) 41-50 (Mass Transport)	MAGNITUDE OF HEAT TRANSFER COEFFICIENT λ(w/m-°c) α_v(w/m²-°c^a) α_w(w/m²-°c) ε $\rho c_p v$(w/°c)	EXPONENT USED WITH TEMP. DIFFERENCE --- a(default = 1.25) --- --- ---	($Q = \lambda A\ \Delta T/l$) ($Q = \alpha_v A\ \Delta T^a$) ($Q = \alpha_w A\ \Delta T$) $Q = \varepsilon\sigma A(T_i^4 - T_j^4)$ ρ = Density (KG/M³), Cp = Specific Heat (W-SEC/KG-°c), v = Flow Rate (M³/SEC)				

*The forced convection coefficient can be calculated internally by the program using one of three options. For options 1 and 3, α_w is calculated by $\alpha_w = \lambda\ Nu/L$ where $Nu = KRe^A Pr^B$ (Re = Reynold's No. = $UL\rho/\eta$, Pr = Prandtl No. = $\eta C_p/\lambda$). For option 2, $\alpha_w = c\eta^D$. The viscosity is a function of temperature for options 2 and 3. To activate this feature, input data for card T7 as shown below and follow immediately with card T7A.

HEAT TRANS. INDEX							
21-30	K	A	B	L (M)	U (M/SEC)	λ (W/m-°C)	(OPTION 1)
21-30	C	D	---	---	---	---	(OPTION 2)
21-30	K	A	B	L (M)	U (M/SEC)	λ (W/m-°C)	(OPTION 3)

DATA CARD T7A Input only if it is desired to calculate forced convection coefficient internally by the program.

Columns 1–10	11–20	21–30	31–40	41–50	51–60	61–70	71–80
F10.0	F10.0	F10.0	F10.0	F10.0	F10.0	F10.0	
η(N-sec/M²)	ρ(KG/M³)	C_p(W-sec/KG-°c)	---	---	---	---	(OPTION 1)
---	---	---	T_L (°c)	η @ T_L (N-sec/M²)	T_H (°c)	η @ T_H (N-sec/M²)	(OPTION 2)
---	ρ(KG/M³)	C_p(W-sec/KG-°c)	T_L (°c)	η @ T_L (N-sec/M²)	T_H (°c)	η @ T_H (K-sec/M²)	(OPTION 3)

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DATA CARD T8

Heat Flow Paths

USE As Many Cards As Needed, Max. 500, Followed By A Blank Card.

Columns 1–10	Columns 11–15	Columns 16–20	Columns 21–30	Columns 31–40	Columns 41–50	Columns 51–60
XX (9–10)	XX (14–15)	XX (19–20)				
I10	I5	I5	F10.0	F10.0	F10.0	F10.0
INDEX (INDAB = [INDEX])	NODE I	NODE J				No. of planets in the stage. Input only if heat transfer involves a planetary subsystem node and a non-planetary node. See section 2.4.8 for detailed instructions.
1 ≤ INDAB ≤ 10	NODE I	NODE J	L_1 (mm)	L_2 (mm)	L_3 (mm)	Conduction between I and J area = $2\pi L_1L_2$. If index < 0 area = L_1L_2. Distance I-J = L_3.
11 ≤ INDAB ≤ 20	NODE I	NODE J	L_1	L_2	BLANK	Natural convection between I and J. Area = $2\pi L_1L_2$. If index < 0 area = L_1L_2.
21 ≤ INDAB ≤ 30	NODE I	NODE J	L_1	L_2	BLANK	Forced convection between I and J. Area as above. If $\eta(t)$, t is t_J.
31 ≤ INDAB ≤ 40	NODE I	NODE J	L_1	L_2	(L_3)	Radiation between I and J. Area as above. For description of L_3, see User's Manual.
41 ≤ INDAB ≤ 50	NODE I	NODE J	Index of fluid Flow at NODE J, 41 ≤ INDEX ≤ 50	BLANK	BLANK	Fluid flow from node I to node J. First index is index of fluid flow at node I. Second index corresponds to fluid flow from I to J.
INDEX = 51	NODE I	NODE J	Stage Number 1 ≤ NO. ≤ 3	Raceway Flag 1. Inner Race Contact 2. Outer Race Contact		

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DATA CARD T9

Node heat capacities, only for transient calculations

USE One Card/Node

Columns 1–10	Columns 11–20	Columns 21–30	Columns 31–40	Columns 41–50	Columns 51–60	Columns 61–70	Columns 71–80
X X I 1 0	F 1 0 . 0		F 1 0 . 0	F 1 0 . 0	F 1 0 . 0		
Node Number	Volume at Node = $L_1 \cdot L_2 \cdot L_3$			Density (KG/M^3)	Specific Heat WS/KG^oC		
If it is given without a sign, rotational symmetry is assumed and the volume is multiplied by 2 π. If it is negative the volume is not changed.	L_1 (mm)	L_2 (mm)	L_3 (mm)				

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APPENDIX B

SAMPLE OUTPUT

"NASA" VERSION

***** P L A N E T S Y S **** TECHNOLOGY DIVISION S K F INDUSTRIES INC. **** P L A N E T S Y S ******

PLANETSYS EXECUTED USING N A S A TRACTION MODEL

THIS PLANETARY SYSTEM CONTAINS 1 STAGE.

STAGE NO. (1) CONTAINS 3 IDENTICAL PLANETS, EACH SUPPORTED BY A SPHERICAL ROLLER BEARING HAVING 2 ROWS OF ROLLERS.

THE MAXIMUM NO. OF MAIN LOOP ITTERATIONS ALLOWED IS 20 AND THE RELATIVE ACCURACY REQUIRED IS .10000-07

***** P L A N E T S Y S **** TECHNOLOGY DIVISION S K F INDUSTRIES INC. **** P L A N E T S Y S ******

PLANETSYS EXECUTED USING N A S A TRACTION MODEL

N O T E S 1) DIMENSIONAL UNITS

ANGLES	DEGREES
DENSITY	POUNDS/CUBIC INCH
ELASTIC MODULI	POUNDS/SQ. INCH
FORCE	POUNDS
KINEMATIC VISC	INCHES SQ./SEC.
LENGTH	INCHES
MASS	POUNDS
PRESSURE	POUNDS/SQ. INCH
SPEEDS	R.P.M.
SURFACE ROUGHNESS	MICROINCHES
TEMPERATURE	DEGREES FAHRENHEIT
THERM. COND	B.T.U./HR-FOOT-DEG F.

2) THE TERM BEARING NUMBER WHICH APPEARS BELOW IS SYNONOMOUS WITH STAGE NUMBER .

3) PLANETARY INVERSION REFERS TO THE SPECIFIC KINEMATIC CONDITION AS DESCRIBED IN THE USERS MANUAL.

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PLANETSYS EXECUTED USING N A S A TRACTION MODEL

***** P L A N E T A R Y D A T A ******

BEARING NUMBER	PITCH DIA OF BEARING	PLANET MASS	PITCH DIA OF GEAR	WIDTH OF PLANET	DIA OF PL. NTRL. AXIS	PLANET MOMENT OF INERTIA	PLANET TOOTH PRESS. ANGLE
1	.24+01	3.440	.395+01	1.550	3.256	.011	24.6

BEARING NUMBER	NUMBER OF PLANETS	POWER THRU STAGE	PLAN. INVERS. (SEE NOTE 3)	SPEED OF SUN	SPEED OF CARRIER	SPEED OF RING	PITCH DIA. OF SUN
1	3	202.200	3	1659.0	355.0	.0	3.0

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PLANETSYS EXECUTED USING N A S A TRACTION MODEL

* * * * * * B E A R I N G D A T A * * * * * *

BEARING NUMBER	BEARING TYPE	NO. OF ROWS	NO. OF ROLLING ELTS. PER ROW	CONTACT ANGLE	DIAMETRAL CLEARANCE
1	SPHERICAL ROLLER BEARING	2	12	15.0	.00230

* * * C A G E D A T A * * *

BEARING NUMBER	CAGE TYPE	CAGE POCKET CLEARANCE	RAIL LAND WIDTH	RAIL LAND DIAMETER	RAIL LAND CLEARANCE	CAGE MASS
1	ROLLING ELEMENT RIDING	.0100	.0000	.000	.000	.04

* * * S T E E L D A T A * * *

BEARING NUMBER	STEEL TYPE, INNER RING	LIFE FACTOR	STEEL TYPE, OUTER RING	LIFE FACTOR
1	M-50 FOR-7	1.000	CARBURIZED 9310	1.000

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PLANETSYS EXECUTED USING N A S A TRACTION MODEL

****** R O L L I N G E L E M E N T D A T A ******

BEARING NO.(1), SPHERICAL ROLLER HEARING

ROLLER DIAMETER	ROLLER TOTAL LENGTH	ROLLERPROFILE RADIUS	NO. OF AXIAL SLICES
.5118	.5320	1.4540	11

OUTER RACEWAY		INNER RACEWAY	
EFFECTIVE LENGTH	OSCULATION	EFFECTIVE LENGTH	OSCULATION
.4520	.9690	.4520	.9690

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PLANETSYS EXECUTED USING N A S A TRACTION MODEL

****** S U R F A C E D A T A ******

BEARING NUMBER	CLA ROUGHNESS OUTER	CLA ROUGHNESS INNER	CLA ROUGHNESS ROLL. ELM.	RMS ASPERITY SLOPE OUTER	RMS ASPERITY SLOPE INNER	RMS ASPERITY SLOPE ROLL. ELM.
1	8.00	8.00	6.00	1.000	1.000	1.000

****** L U B R I C A N T D A T A ******

BEARING NUMBER	DESIGNATION	KINEMATIC VISCOSITY (37.78 C)	KINEMATIC VISCOSITY (98.89 C)	DENSITY AT (15.56 C)	THERMAL EXPAN. COEFFICIENT	THERMAL CONDUCTIVITY
1	MIL-L-23699	.04	.01	.04	.41-03	.088

****** L U B R I C A T I O N A N D F R I C T I O N D A T A ******

BEARING NUMBER	PERCENT LUBE IN CAVITY	FILM REPLENISHMENT LAYER THICKNESS (ROLL.ELM. + RACEWAY) OUTER	FILM REPLENISHMENT LAYER THICKNESS (ROLL.ELM. + RACEWAY) INNER	ASPERITY FRICTION COEFFICIENT
1	1.00	.4000-04	.2000-04	.10

GIVEN TEMPERATURES (DEG C)

BRG	SHAFT	I. RING	I. RACE	ROLL EL.	O.RACE	PLANET	CAGE	BULK
1	121.00	121.00	121.00	121.00	121.00	121.00	121.00	121.00

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PLANETSYS EXECUTED USING N A S A TRACTION MODEL

B E A R I N G S Y S T E M O U T P U T ENGLISH UNITS

STAGE NO.	PLANET GEAR SPEED	SUN GEAR SPEED	RING GEAR SPEED	PLANET ORBITAL SPEED	ROLLER ORBITAL SPEED	ROLLER ROTATIONAL SPEED
1	-.100+04	.166+04	.000	361.	-591.	-.232+04

STAGE NO.	TANGENTIAL GEAR LOAD*	RADIAL GEAR LOAD*	GEAR TOOTH TORQUE*	PLANET GEAR INERTIA LOAD
1	.168+04	369.	584.	44.6

* LOAD AT ONE GEAR TOOTH.

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PLANETSYS EXECUTED USING N A S A TRACTION MODEL

STG.	FATIGUE LIFE(HOURS) OUTER RACE	INNER RACE	SINGLE ROW BEARING	DOUBLE ROW BEARING	H / SIGMA OUTER RACE	INNER RACE
1	.27+05	.27+04	.25+04	.14+04	.05	.04

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STG.	LUBE-LIFE FACTOR OUTER RACE	INNER RACE	MATERIAL FACTOR OUTER RACE	INNER RACE
1	.210	.210	1.000	1.000

TEMPERATURES RELEVANT TO BEARING PERFORMANCE (DEGREES FAHRENHEIT)

BRG.	SHAFT	I. RING	I. RACE	I. FLNG.	ROLL. EL	O. FLNG.	O. RACE	O. RING	BULK LUBE
1	250.	250.	250.	250.	250.	250.	250.	250.	250.

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PLANETSYS EXECUTED USING N A S A TRACTION MODEL

B E A R I N G S Y S T E M O U T P U T ENGLISH UNITS

FRICTIONAL HEAT GENERATION RATE (BTU/HR) AND FRICTION TORQUE (LB-IN)

STG.	O. RACE	I. RACE	R.E. DRAG	R.E.-CAGE	CAGE-LAND	TOTAL	TORQUE
1	96.2	333.	0.000	9.25	0.000	439.	10.9

EHD FILM THICKNESS, FILM REDUCTION FACTORS AND HEAT CONDUCTIVITY DATA FOR THE OUTER AND INNER RACEWAYS RESPECTIVELY

STG.	FILM (MICRO IN)		STARVATION FACTOR		THERMAL FACTOR		MENISCUS DIST. (IN)		CONDUCTIVITY (B/HR/DEG.F)	
1	.683	.507	1.00	1.00	.667	.660	9.282-02	5.434-02	25.3	22.8

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PLANETSYS EXECUTED USING N A S A TRACTION MODEL

B E A R I N G S Y S T E M O U T P U T ENGLISH UNITS

LUBRICANT TEMPERATURES AND PHYSICAL PROPERTIES

	LOCATION	TEMPERATURES (DEGREES F.)	DENSITY (LB/IN3)	VISCOSITY KIN. (IN2/SEC)	VISCOSITY DYN. (LB-SEC/IN2)	PRESSURE VISCOSITY COEFFICIENT (1/PSI)
BRG. 1	OUTER	249.800	.3366-01	.5415-02	.4720-06	.7797-04
	INNER	249.800	.3366-01	.5415-02	.4720-06	.7797-04
	BULK	249.800	.3366-01	.5415-02	.4720-06	.7797-04

C A G E D A T A ENGLISH UNITS

CAGE RAIL - RING LAND DATA CAGE SPEED DATA

BRG.	TORQUE (IN-LB)	HEAT RATE (BTU/HR)	SEP.FORCE (POUNDS)	ECCENTRICITY RATIO	CALCULATED SPEED (RAD/SEC)	CALCULATED SPEED (RPM)
1	0.000	0.000	4.000-02	0.000	-61.9	-591.

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C-2

***** P L A N E T S Y S **** TECHNOLOGY DIVISION S K F INDUSTRIES INC. **** P L A N E T S Y S ******

PLANETSYS EXECUTED USING N A S A TRACTION MODEL

ROLLING ELEMENT OUTPUT FOR BEARING NUMBER 1 ENGLISH UNITS

AZUMITH ANGLE (DEG)	NORMAL FORCES CASE	ROLLER END	OUTER	INNER	HZ STRESS OUTER	INNER	O. RING DEFL.	CENTRIFUGAL FORCE
15.000	.104	.114	.45[illegible]	.031	.137+05	.436+04	.11-02	.44+00
45.000	.283	.088	.351	.023	.120+05	.382+04	-.39-06	.34+00
75.000	.386	.041	.165	.011	.821+04	.262+04	-.10-02	.16+00
105.000	.386	-.012	.000	.046	.000	.335+04	-.10-02	-.48-01
135.000	.283	-.059	.000	.219	.000	.117+05	.33-04	-.23+00
165.000	.104	-.086	.000	.319	.000	.141+05	.11-02	-.33+00
195.000	-.104	-.086	242.210	242.529	.143+06	.176+06	.89-03	-.33+00
225.000	-.283	-.059	508.327	508.547	.185+06	.228+06	-.11-03	-.23+00
255.000	-.386	-.012	468.933	468.979	.179+06	.221+06	-.90-03	-.48-01
285.000	-.386	.041	468.832	468.678	.179+06	.221+06	-.89-03	.16+00
315.000	-.283	.088	502.590	502.262	.184+06	.227+06	-.86-04	.34+00
345.000	-.104	.114	221.663	221.236	.138+06	.170+06	.91-03	.44+00

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PLANETSYS EXECUTED USING N A S A TRACTION MODEL

CONTACT ELLIPSE SIZE FOR STAGE NO. 1 ENGLISH UNITS

AZUMITH ANGLE (DEG)	OUTER RING SEMI-MAJOR	OUTER RING SEMI-MINOR	INNER RING SEMI-MAJOR	INNER RING SEMI-MINOR
15.000	.1657-01	.7062-03	.6714-02	.2406-03
45.000	.1516-01	.6461-03	.6143-02	.2201-03
75.000	.1173-01	.5025-03	.4778-02	.1712-03
105.000	.0000	.0000	.7693-02	.2[illegible]-03
135.000	.0000	.0000	.1294-01	.4635-03
165.000	.0000	.0000	.1466-01	.5253-03
195.000	.1340+00	.5712-02	.1338+00	.4792-02
225.000	.1715+00	.7313-02	.1712+00	.6134-02
255.000	.1670+00	.7119-02	.1667+00	.5971-02
285.000	.1670+00	.7118-02	.1666+00	.5969-02
315.000	.1703+00	.7285-02	.1705+00	.6109-02
345.000	.1301+00	.5545-02	.1297+00	.4648-02

A (*) FOLLOWING AN ELLIPSE DIMENSION INDICATES A CONTACT LENGTH GREATER THAN THE EFFECTIVE LENGTH
A (**) FOLLOWING AN ELLIPSE DIMENSION INDICATES A CONTACT LENGTH GREATER THAN 1.5 TIMES THE EFFECTIVE LENGTH

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APPENDIX C

SAMPLE OUTPUT

"SKF" VERSION

***** P L A N E T S Y S **** TECHNOLOGY DIVISION S K F INDUSTRIES INC. **** P L A N E T S Y S ******

PLANETSYS EXECUTED USING SKF TRACTION MODEL

THIS PLANETARY SYSTEM CONTAINS 1 STAGE.

STAGE NO. (1) CONTAINS 3 IDENTICAL PLANETS, EACH SUPPORTED BY A SPHERICAL ROLLER BEARING HAVING 2 ROWS OF ROLLERS.

THE MAXIMUM NO. OF MAIN LOOP ITTERATIONS ALLOWED IS 20 AND THE RELATIVE ACCURACY REQUIRED IS .10000-07

***** P L A N E T S Y S **** TECHNOLOGY DIVISION S K F INDUSTRIES INC. **** P L A N E T S Y S ******

PLANETSYS EXECUTED USING SKF TRACTION MODEL

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N O T E S 1)DIMENSIONAL UNITS

ANGLES..........................DEGREES
DENSITY...............POUNDS/CUBIC INCH
ELASTIC MODULI...........POUNDS/SQ. INCH
FORCE...........................POUNDS
KINEMATIC VISC...........INCHES SQ./SEC.
LENGTH..........................INCHES
MASS............................POUNDS
PRESSURE................POUNDS/SQ. INCH
SPEEDS..........................R.P.M.
SURFACE ROUGHNESS...........MICROINCHES
TEMPERATURE...........DEGREES FAHRENHEIT
THERM. COND........B.T.U./HR-FOOT-DEG F.

2)THE TERM BEARING NUMBER WHICH APPEARS BELOW IS SYNONOMOUS WITH STAGE NUMBER .

3) PLANETARY INVERSION REFERS TO THE SPECIFIC KINEMATIC CONDITION AS DESCRIBED IN THE USERS MANUAL.

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PLANETSYS EXECUTED USING SKF TRACTION MODEL

****** P L A N E T A R Y D A T A ******

BEARING NUMBER	PITCH DIA OF BEARING	PLANET MASS	PITCH DIA OF GEAR	WIDTH OF PLANET	DIA OF PL. NTRL. AXIS	PLANET MOMENT OF INERTIA	PLANET TOOTH PRESS. ANGLE
1	.24+01	3.440	.395+01	1.550	3.256	.011	24.6

BEARING NUMBER	NUMBER OF PLANETS	POWER THRU STAGE	PLAN. INVERS. (SEE NOTE 3)	SPEED OF SUN	SPEED OF CARRIER	SPEED OF RING	PITCH DIA. OF SUN
1	3	202.200	3	1659.0	355.0	.0	3.0

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PLANETSYS EXECUTED USING SKF TRACTION MODEL

***** B E A R I N G D A T A *****

BEARING NUMBER	BEARING TYPE	NO. OF ROWS	NO. OF ROLLING ELTS. PER ROW	CONTACT ANGLE	DIAMETRAL CLEARANCE
1	SPHERICAL ROLLER BEARING	2	12	15.0	.00230

*** C A G E D A T A ***

BEARING NUMBER	CAGE TYPE	CAGE POCKET CLEARANCE	RAIL LAND WIDTH	RAIL LAND DIAMETER	RAIL LAND CLEARANCE	CAGE MASS
1	ROLLING ELEMENT RIDING	.0100	.0000	.000	.000	.04

*** S T E E L D A T A ***

BEARING NUMBER	STEEL TYPE, INNER RING	LIFE FACTOR	STEEL TYPE, OUTER RING	LIFE FACTOR
1	M-50 FOR-7	1.000	CARBURIZED 9310	1.000

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PLANETSYS EXECUTED USING SKF TRACTION MODEL

******ROLLING ELEMENT DATA******

BEARING NO.(1), SPHERICAL ROLLER BEARING

ROLLER DIAMETER	ROLLER TOTAL LENGTH	ROLLERPROFILE RADIUS	NO. OF AXIAL SLICES
.5118	.5320	1.4540	11

OUTER RACEWAY		INNER RACEWAY	
EFFECTIVE LENGTH	OSCULATION	EFFECTIVE LENGTH	OSCULATION
.4520	.9690	.4520	.9690

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PLANETSYS EXECUTED USING SKF TRACTION MODEL

***** S U R F A C E D A T A *****

BEARING NUMBER	CLA ROUGHNESS OUTER	CLA ROUGHNESS INNER	CLA ROUGHNESS ROLL. ELM.	RMS ASPERITY SLOPE OUTER	RMS ASPERITY SLOPE INNER	RMS ASPERITY SLOPE ROLL. ELM.
1	8.00	8.00	6.00	1.000	1.000	1.000

***** L U B R I C A N T D A T A *****

BEARING NUMBER	DESIGNATION	KINEMATIC VISCOSITY (37.78 C)	KINEMATIC VISCOSITY (98.89 C)	DENSITY AT (15.56 C)	THERMAL EXPAN. COEFFICIENT	THERMAL CONDUCTIVITY
1	MIL-L-23699	.04	.01	.09	.41-03	.088

***** L U B R I C A T I O N A N D F R I C T I O N D A T A *****

BEARING NUMBER	PERCENT LUBE IN CAVITY	FILM REPLENISHMENT LAYER THICKNESS (ROLL.ELM. + RACEWAY) OUTER	FILM REPLENISHMENT LAYER THICKNESS (ROLL.ELM. + RACEWAY) INNER	ASPERITY FRICTION COEFFICIENT
1	1.00	.4000-04	.2000-04	.10

GIVEN TEMPERATURES (DEG C)

BRG	SHAFT	I. RING	I. RACE	ROLL EL.	O.RACE	PLANET	CAGE	BULK
1	121.00	121.30	121.00	121.00	121.00	121.00	121.00	121.00

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PLANETSYS EXECUTED USING SKF TRACTION MODEL

B E A R I N G   S Y S T E M   O U T P U T   ENGLISH UNITS

STAGE NO.	PLANET GEAR SPEED	SUN GEAR SPEED	RING GEAR SPEED	PLANET ORBITAL SPEED	ROLLER ORBITAL SPEED	ROLLER ROTATIONAL SPEED
1	-.100+04	.166+04	.000	361.	-591.	-.232+04

STAGE NO.	TANGENTIAL GEAR LOAD*	RADIAL GEAR LOAD*	GEAR TOOTH TORQUE*	PLANET GEAR INERTIA LOAD
1	.168+04	769.	584.	44.6

* LOAD AT ONE GEAR TOOTH.

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PLANETSYS EXECUTED USING SKF TRACTION MODEL

	FATIGUE LIFE(HOURS)				H / SIGMA	
STG.	OUTER RACE	INNER RACE	SINGLE ROW BEARING	DOUBLE ROW BEARING	OUTER RACE	INNER RACE
1	.27+05	.27+04	.25+04	.14+04	.07	.06

	LUBE-LIFE FACTOR		MATERIAL FACTOR	
STG.	OUTER RACE	INNER RACE	OUTER RACE	INNER RACE
1	.210	.210	1.000	1.000

TEMPERATURES RELEVANT TO BEARING PERFORMANCE (DEGREES FAHRENHEIT)

BRG.	SHAFT	I. RING	I. RACE	I. FLNG.	ROLL. EL	O. FLNG.	O. RACE	O. RING	BULK LUBE
1	250.	250.	250.	250.	250.	250.	250.	250.	250.

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PLANETSYS EXECUTED USING SKF TRACTION MODEL

B E A R I N G   S Y S T E M   O U T P U T   ENGLISH UNITS

FRICTIONAL HEAT GENERATION RATE (BTU/HR) AND FRICTION TORQUE (LB-IN)

STG.	O. RACE	I. RACE	R.E. DRAG	R.E.-CAGE	CAGE-LAND	TOTAL	TORQUE
1	140.	484.	0.000	9.25	0.000	633.	15.7

EHD FILM THICKNESS, FILM REDUCTION FACTORS AND HEAT CONDUCTIVITY DATA FOR THE OUTER AND INNER RACEWAYS RESPECTIVELY

STG.	FILM (MICRO IN)		STARVATION FACTOR		THERMAL FACTOR		MENISCUS DIST. (IN)		CONDUCTIVITY (B/HR/DEG.F)	
1	.926	.753	1.00	1.00	.667	.660	9.282-02	5.434-02	24.8	22.3

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PLANETSYS EXECUTED USING SKF TRACTION MODEL

B E A R I N G   S Y S T E M   O U T P U T   ENGLISH UNITS

LUBRICANT TEMPERATURES AND PHYSICAL PROPERTIES

	LOCATION	TEMPERATURES (DEGREES F.)	DENSITY (LB/IN3)	VISCOSITY KIN. (IN2/SEC)	VISCOSITY DYN. (LB-SEC/IN2)	PRESSURE VISCOSITY COEFFICIENT (1/PSI)
BRG. 1	OUTER	249.800	.3366-01	.5415-02	.4720-06	.7797-04
	INNER	249.800	.3366-01	.5415-02	.4720-06	.7797-04
	BULK	249.800	.3366-01	.5415-02	.4720-06	.7797-04

C A G E   D A T A   ENGLISH UNITS

	CAGE RAIL - RING LAND DATA				CAGE SPEED DATA	
BRG.	TORQUE (IN-LB)	HEAT RATE (BTU/HR)	SEP.FORCE (POUNDS)	ECCENTRICITY RATIO	CALCULATED SPEED (RAD/SEC)	(RPM)
1	0.000	0.000	4.000-02	0.000	-61.9	-591.

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PLANETSYS EXECUTED USING SKF TRACTION MODEL

R O L L I N G E L E M E N T O U T P U T FOR BEARING NUMBER 1 ENGLISH UNITS

AZUMITH ANGLE (DEG)	CAGE	NORMAL FORCES ROLLER END	OUTER	INNER	HZ STRESS OUTER	INNER	O. RING DEFL.	CENTRIFUGAL FORCE
15.000	.104	.114	.458	.031	.137+05	.436+04	.11-02	.44+00
45.000	.283	.088	.351	.023	.120+05	.382+04	-.39-06	.34+00
75.000	.386	.041	.165	.011	.821+04	.262+04	-.10-02	.16+00
105.000	.386	-.012	.000	.046	.000	.535+04	-.10-02	-.48-01
135.000	.283	-.059	.000	.219	.000	.117+05	.33-04	-.23+00
165.000	.104	-.086	.000	.319	.000	.141+05	.11-02	-.33+00
195.000	-.104	-.086	242.210	242.529	.143+06	.176+06	.89-03	-.33+00
225.000	-.283	-.059	508.327	508.547	.185+06	.228+06	-.11-03	-.23+00
255.000	-.386	-.012	468.933	468.979	.179+06	.221+06	-.90-03	-.48-01
285.000	-.386	.041	468.832	468.678	.179+06	.221+06	-.89-03	.16+00
315.000	-.283	.088	502.590	502.262	.184+06	.227+06	-.86-04	.34+00
345.000	-.104	.114	221.663	221.236	.138+06	.170+06	.91-03	.44+00

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PLANETSYS EXECUTED USING SKF TRACTION MODEL

CONTACT ELLIPSE SIZE FOR STAGE NO. 1 ENGLISH UNITS

AZIMUTH	OUTER RING		INNER RING	
ANGLE (DEG)	SEMI-MAJOR	SEMI-MINOR	SEMI-MAJOR	SEMI-MINOR
15.000	.1657-01	.7062-03	.6714-02	.2486-03
45.000	.1515-01	.6461-03	.6143-02	.2281-03
75.000	.1170-01	.5025-03	.4778-02	.1712-03
105.000	.0000	.0000	.7693-02	.2756-03
135.000	.0000	.0000	.1294-01	.4635-03
165.000	.0000	.0000	.1466-01	.5253-03
175.000	.1340+00	.5712-02	.1338+00	.4792-02
225.000	.1715+00	.7313-02	.1712+00	.6134-02
255.000	.1670+00	.7119-02	.1667+00	.5971-02
285.000	.1570+00	.7118-02	.1666+00	.5969-02
315.000	.1700+00	.7285-02	.1705+00	.6109-02
345.000	.1301+00	.5545-02	.1297+00	.4648-02

A (*) FOLLOWING AN ELLIPSE DIMENSION INDICATES A CONTACT LENGTH GREATER THAN THE EFFECTIVE LENGTH
A (**) FOLLOWING AN ELLIPSE DIMENSION INDICATES A CONTACT LENGTH GREATER THAN 1.5 TIMES THE EFFECTIVE LENGTH

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APPENDIX D

SKF COMPUTER PROGRAM

"PLANETSYS" FLOW CHART

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"PLANETSYS" FLOWCHART

```
PLASYS ( MAIN PROGRAM )
  SKFPL
    DATIN
      PROPST
      LUPROP
      LUBCON
      TYPE
    DATOUT
      TITLE
      FITDAT
    CNVRT
```

Read and write planet gear and bearing input data. Convert input units to the English system if required.

```
  TEMPIN
    INDUM
    RWHTC
    RWG
    RWHC
    TMAP
```

Read & write steady state or transient thermal input data.

```
  CONSPL
    CONST
      IBDSET
      CONCY
      CONSPH
      CONSPD
      CONLOD
      CONPL
      CONERT
```

Calculate planet bearing system related loads, speeds and constants.

```
  ALLTPL
```

Begin in the system solution.

```
    PLABRG
```

Begin the calculation of the planet bearing load distribution and heat generation rates.

```
      FIT
          INTFIT
            SIMEQ
```

Calculate the bearing diametral clearance if required.

```
      GUESPL
      PREPPL
      SOLV13
          INSOLV
          EQS = PLNTEQ
            RACLOD
          PARDER
            EQS = PLNTEQ
          SIMQ
          EQCHEK
            EQS = PLNTEQ
            ERWRIT
            DAMPCO
            ERCHEK
```

Prepare, then solve planet gear equilibrium. This will yield the load distribution among the rolling elements.

```
      IRLOAD
```

Calculate the rolling element-inner ring loads.

```
      VISCO2
          ALPHAO
      DRAGNO
      STCON
```

Calculate lubricant temperature related constants.

```
      BRGDAT
          DISLOD
          ROLSPD
          FMIXPL
            SPEEDS
            THERFI
            STARFC
            HDFRIC
            ASLOAD
            SPEEDS
            EHDSKF
              FRICTN
```

Calculate rolling element raceway EHD conditions, and frictional heat generation rates.

(CONTINUED)

"PLANETSYS" FLOWCHART (continued)

```
| | | | |   CAGE
| | | | |    | CGROL                     Calculate the cage-
| | | | |    |  | WEBWET                 rolling element and cage-
| | | | |    |  | WEBDRY                 land frictional heat
| | | | |    |  | ENDWET                 generation rates.
| | | | |    |  | ENDDRY
| | | | |    | LANDHT
| | | | ---------------------
| | | | |   DRAGHT                       Calculate the rolling
| | | | |   FILLGB                       element drag heat, the
| | | | |   PLLIFE                       bearing fatigue life and
| | | | |    |FLMFAC                     rolling element raceway
| | | | |   PLHUT                        heat transfer coeffi-
| | | ------------------------           cients.
| | | DELIV3
| | |  | TITLE                           Write output data from
| | |  | RITE                            the planetary system
| | |  | REQUT3                          solution.
| | |  |  | RITE2
| | | ------------------------
| | | FILLGT
| | | SOLV13                             Solve the temperature
| | |  | EQS = NET                       distribution for steady
| | |  |  | NETEET                       state equilibrium and
| | | DELIV3                             write results.
| | | TMAP
| | | ------------------------
| | | STEPMA
| | |  | NET                             Solve the temperature
| | |  |  | NETEET                       distribution for the
| | | DELIV3                             transient time step and
| | | NET                                write results.
| | | TMAP
| | | ------------------------
| | | DELIV3                             Write all final planetary
| | | TMAP                               system performance and
| | | ------------------------           temperature distribution
| | |                                    results.
```

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APPENDIX E

HEAT TRANSFER INFORMATION

E.1 BACKGROUND

The temperature portion of the Program is designed to produce temperature maps for an axisymmetric mechanical system of any geometrical shape. The mechanical system is first approximated by an equivalent system comprising a number of elements of simple geometries. Each element is then represented by a node point having either a known or an unknown temperature. The environment surrounding the system is also represented by one or more nodes. With the node points properly selected, the heat balance equations can be set up accordingly for the nodes of unknown temperature. These equations become non-linear when there is free convection and/or radiation between two or more of the node points considered. The problem is therefore reduced to solving a set of linear and/or non-linear equations for the same number of unknown nodal temperatures. It is obvious that the success of the approach depends largely on the physical subdivision of the system. If the subdivision is too fine, there will be a large number of equations to be solved; on the other hand, if the subdivision is too crude, the results may not be reliable.

In a system consisting of rolling bearings, for the sake of simplicity, the elements considered are usually axially symmetrical, e.g., each of the bearing rings can be taken as an element of uniform temperature. For an element which is not axially symmetrical, its temperature is also assumed to be uniform and its presence is assumed not to distort the uniformity in temperature of a neighboring element which is axially symmetrical. That is, the non-symmetrical element is represented by an equivalent axially symmetrical element with approximately the same surface area and material volume. This kind of approximation may seem to be somewhat unrealistic, but with properly devised equivalent systems, it can be used to solve complicated problems with results satisfying most of the important engineering requirements.

The computer program can solve the heat-balance equations for either the steady state or the transient state conditions and produce temperature maps for the mechanical system when the input data are properly prepared.

E.2 BASIC EQUATIONS

E.2.1 Heat Conduction

The rate of heat flow $q_{ci,j}$(W) that is conducted from node i to node j may be represented by,

$$q_{ci,j} = \frac{\lambda_{ij} A_{ij}}{L_{ij}} (t_i - t_j)$$

t_i and t_j are the temperatures at i and j, respectively, $A_{i,j}$ the area normal to the heat flow (m^2), L_{ij} the distance (m) and λ_{ij} the thermal conductivity between i and j, (W/M°C).

Assuming that the structure between point i and j is composed of different materials, on equivalent heat conductivity may be calculated as follows:

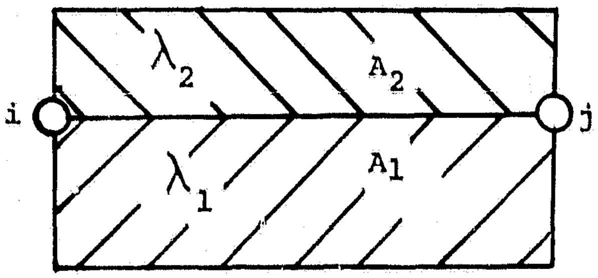


Fig. E-1

$$\lambda_{ij} = \frac{\lambda_1 A_1 + \lambda_2 A_2}{A_{ij}}$$

$$A_{ij} = A_1 + A_2$$

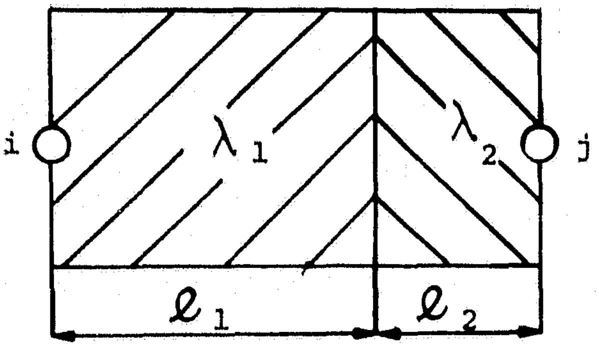


Fig. E-2

$$\lambda_{ij} = \frac{\ell_{ij}}{\ell_1/\lambda_1 + \ell_2/\lambda_2}$$

$$\ell_{ij} = \ell_1 + \ell_2$$

The calculation of the areas will be discussed in Section E.2.5.

E.2.2 CONVECTION

The rate of heat flow that is transferred between a solid structure and air by free convection may be expressed by

$$q_{vi,j} = \alpha_{i,j} A_{i,j} |t_i - t_j|^{1.25} \cdot \text{SIGN}(t_i - t_j)$$

where

$$\text{SIGN} = \begin{cases} 1, \text{ if } (t_i - t_j) \geq 0 \\ -1, \text{ if } (t_i - t_j) < 0 \end{cases}$$

in which

$$\alpha_{ij} = \begin{cases} 2.5 \cdot 10^{-2}\ W/m^2 - (degC)^{1.25} & \text{for hot surfaces facing upward and cold surfaces facing downward} \\ 1.4 \cdot 10^{-2}\ W/m^2 - (degC)^{1.25} & \text{for hot surfaces facing downward and cold surfaces facing upward} \\ 1.8 \cdot 10^{-2}\ W/m^2 - (degC)^{1.25} & \text{for vertical surfaces} \end{cases}$$

For other special conditions, α_{ij} must be estimated by referring to heat transfer literature.

The rate of heat flow that is transferred between a solid structure and a fluid by forced convection my be expressed by

$$q_{wi,j} = \alpha_{i,j} A_{i,j} (t_i - t_j)$$

in which $\alpha_{i,j}$ is the heat transfer coefficient.

Now, with $\alpha = \alpha_{ij}$, introduce the Nusselt number

$$N_u = \frac{\alpha L}{\lambda}$$

the Reynolds number

$$R_e = \frac{UL}{\nu}$$

and the Prandtl number

$$P_r = \frac{\rho\ \nu\ c_p}{\lambda}$$

where

L is a characteristic length which is equal to the diameter in the case of a cylindrical surface and is equal to the plate length in case of a flat surface (m)

U is a characteristic velocity which is equal to the difference between the fluid velocity at some distance from the surface and the surface velocity (m/sec)

λ is the fluid thermal conductivity (W/M^oC)

ν is the fluid kinematic viscosity (M^2/sec)

ρ is the fluid density (kg/m^3)

C_p is the fluid specific heat (J/kg^oC)

For given values of R_e and P_r the Nusselt number N_u and thus the heat transfer coefficent may be estimated from one of the following expressions:

Laminar flow along a flat place: $R_e < 2300$

$$N_u = 0.323 \sqrt{R_e} \cdot \sqrt[3]{P_r}$$

Laminar flow of a liquid in a pipe:

$$N_u = 1.36 \sqrt[3]{R_e \cdot P_r(\frac{D}{L})}$$

where D is the pipe diameter and L the pipe length

Turbulent flow of a liquid in a pipe:

$$N_u = 0.027 \cdot R_e^{0.8} \cdot \sqrt[3]{P_r}$$

Gas flow inside and outside a tube:

$$N_u = 0.3R_e^{.57}$$

Liquid flow outside a tube:

$$N_u = 0.6\ R_e^{.5} \cdot P_r^{0.31}$$

Forced free convection from the outer surface of a rotating shaft

$$N_u = 0.11\ [0.5\ R_e^{\ 2} \cdot P_r]^{0.35}$$

where the Reynolds number R_e is developed by the shaft rotation.

$$R_e = \frac{\omega \pi D^2}{\nu}$$

in which ω is the angular velocity (rad/sec)
D is the shaft diameter (m)

The average coefficient of forced convection to the lubricating oil within a rolling contact bearing may be approximated by,

$$\alpha = 0.0986 \left\{ \frac{N}{\nu} \left[1 \pm \frac{D \cos \ (\beta)}{d_m} \right] \right\}^{1/2} \lambda (P_r)^{1/3}$$

using + for outer ring rotation
- for inner ring rotation

in which N is the bearing operating speed (rad/sec)
D is the diameter of the rolling elements (mm)
d_m is the bearing pitch diameter (mm)
B is the bearing contact angle (degrees)

E.2.3 FLUID FLOW

The rate of heat flow that is transferred from fluid node i to fluid node j by fluid flow is

$$q_{fi,j} = \rho \dot{V}_{ij} \ C_p \ (t_i - t_j)$$

$\dot{V}_{ij}$ is the volume rate of flow from i to j. It must be observed that the continuity of mass requires the following equation to be satisfied

$$\Sigma \dot{V}_{ij} = 0$$

provided the fluid density is constant. The summation should be extended over all nodes i within the fluid which have heat exchange with node j by fluid flow.

E.2.4 HEAT RADIATION

The rate of heat flow that is radiated to node j from node i is expressed by

$$q_{Ri,j} = \delta_{i,j} \ \{(t_i + 273)^4 - (t_j + 273)^4\}$$

where

t_j = Temperature of node j in ^oC

t_i = Temperature of node i in ^oC

and the value of the coefficient $\delta_{i,j}$ depends on the geometry and the emissivity or the absorptivity of the bodies involved.

For radiation between large, parallel and adjacent surfaces of equal area, $A_{i,j}$ and emissivity, $\varepsilon_{i,j}$, $\delta_{i,j}$ is obtained from the equation

$$\delta_{i,j} = \varepsilon_{i,j} \, \sigma A_{i,j}$$

where σ, the Stefan -Boltzmann constant, is

$$\sigma = 5.76 \cdot 10^{-8} \; W/m^2 \, /(degK)^4$$

For radiation between concentric spheres and coaxial cylinders of equal emissivity, $\varepsilon_{i,j}$, $\delta_{i,j}$ is given by the equation

$$\delta_{i,j} = \frac{\varepsilon_{i,j} \, \sigma A_{i,j}}{1 + (1-\varepsilon_{i,j}) \frac{A_{i,j}}{A^*_{i,j}}}$$

where σ is as above, $A_{i,j}$ is the area of the enclosed body and $A^*_{i,j}$ is the area of the surrounding body, i.e. $A_{i,j} < A^*_{i,j}$.

Expressions for $\delta_{i,j}$ that are valid for more complicated geometries or for different emissivities may be found in the heat transfer literature.

E.2.5 CALCULATION OF AREAS

In the case of conduction heat transfer in the axial direction $A_{i,j}$ is given by the equation (Fig. E-3)

$$A_{i,j} = 2\pi \, r_m \cdot \Delta r$$

Referring to the input instructions, Section 2.4.8

$$L_1 = r_m = \frac{r_1 + r_2}{2}$$

$$L_2 = \Delta r = r_2 - r_1$$

In the case of heat transfer in the radial direction, $A_{i,j}$ is obtained from the expression

$$A_{i,j} = 2\pi\, r_m \cdot H;\; L_1 = r_m;\; L_2 = H$$

and similarly for the radiation term above (Figure E-4 (c))

$$A^*_{i,j} = 2\pi\, r^*_m H$$

$$L_3 = r^*_m$$

$$L_2 = H$$

in which H is the length of the cylindrical surface; where heat is conducted between i and j, r_m is given by the same equation as above (Fig. E-4 (a)); where heat is convected between i and j, r_m is the radius of the cylindrical surface (Fig. E - 4 (b)); where heat is radiated between i and j, r_m is the radius of the enclosed cylindrical surface and r_{m*} the radius of the surrounding cylindrical surface (Fig. E-4 (c))

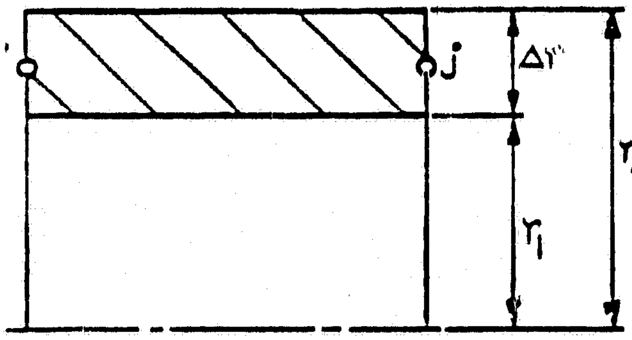


Fig. E-3

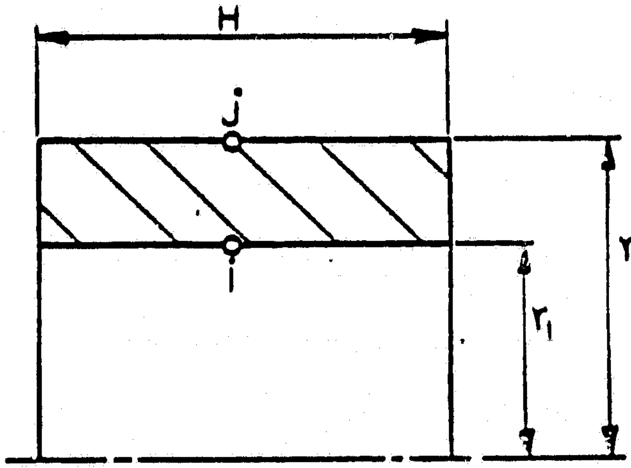


Fig. E-4(a)

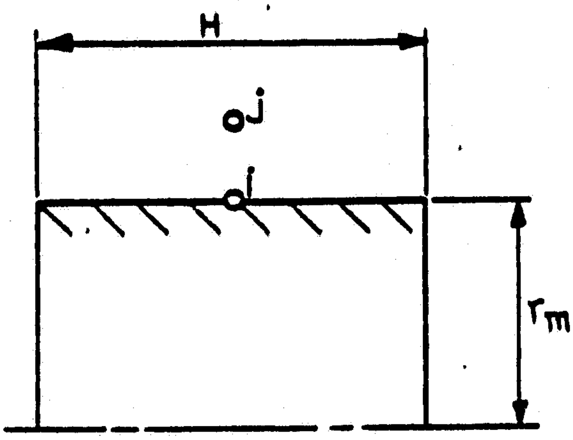


Fig. E -4(b)

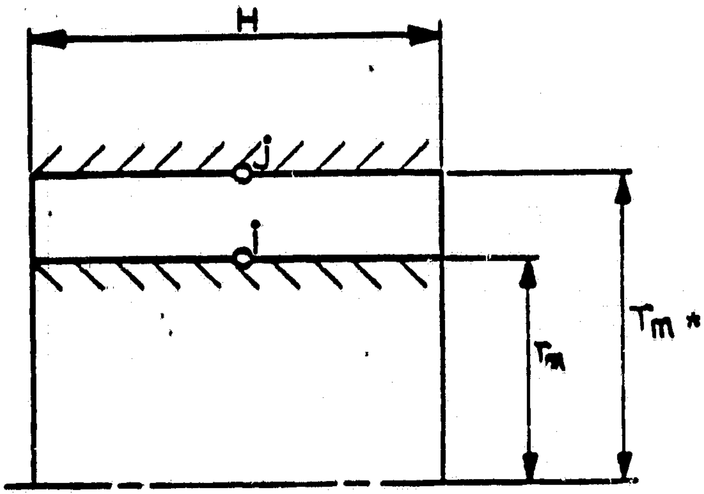


Fig. E-4(c)

E.3.1. TRANSIENT ANALYSIS

For the transient analysis all of the data pertaining to the node to node heat transfer coefficients must be provided by the input. Additionally, the volume and the specific heat at each node is required.

PART 2: EQUATIONS USED IN TEMPERATURE CALCULATIONS

E.4 Temperature Calculations

Subsequent to each calculation of bearing generated heat rates, either the steady state or transient temperature mapping solution scheme may be executed. This set of sequential calculations is terminated as follows:

1. For the steady state case: when each system temperature is within EP1 °Centrigrade of its previously predicted value, EP1 is specified by the user. If it is zero or left blank, a default value of 1° Centigrade is used. This criteria implies that the steady state equilibrium conditions had been reached.

2. The transient calculation terminates when the user specified time up is reached or when one of the system temperatures exceeds 600°C.

E.4.1. Steady State Temperature Map

The mechanical structure to be analyzed is thought of as divided into a number of elements or nodes, each represented by a temperature. The net heat flow to node i from the surrounding nodes j, plus the heat generated at node i, must numerically equal zero. This is true for each node i, i going from 1 to n, n being the number of unknown temperatures.

After each calculation of bearing generated heat, which results from a solution of the planet bearing system portion of the program, a set of system temperatures is determined which satisfy the system of equations:

$$q_i = q_{oi} + q_{gi} = 0 \text{ for all temperature nodes } i \qquad (E.1)$$

where q_{oi} is the heat flow from all neighboring nodes to node i

q_{gi} is the heat generated at node i. These values may be input or calculated by the planet bearing program as bearing frictional heat

This scheme is solved with a modified Newton-Raphson method which successfully terminates when either of two conditions are met:

$$\frac{\Delta t_i}{t_i} \leq EP2 \text{ for all nodes } i \qquad (E.2)$$

Where: Δt represents the Newton-Raphson correction to the temperature t at a given iteration such that, $t_{N+1} = t_N + \Delta t$ and N + 1, and N, refer to the next and current iteration respectively.

EP2 is a user specified constant. If EP2 is left blank or set to zero (0) a default value of 0.001 is used.

A second convergence criterion dependent upon EP2 is also used. In the system of equations, $q_{oi} + q_{gi} = 0$ for all nodes i, absolute convergence would be otained if the right hand side (EQ) in fact reduced to zero (0). Usually a small residue remains at each node, such that $(q_{oi} + q_{gi}) = (EQ)i$.

The second convergence criterion is satisfied if

$$\left[\sum_{i=1}^{n} \frac{(EQ)_i^2}{n}\right]^{1/2} \leq 100 \text{ X } EP2 \qquad (E.3)$$

where n = number of equations in thermal solution
= number of unknown temperatures

E.4.2 Transient Temperatures

In the transient case the net heat q_i transferred to a node i heats the element. It is thus necessary for heat balance at node i that the following equations are satisfied.

$$\rho i \; Cp_i \; V_i \frac{dt_i}{dT_i} = q_i \qquad (E.4)$$

where ρ = density
C_p = specific heat
V = volume of the element
t = temperature
T = time

The temperatures, t_{oi}, at the time of initiation $T = T_s$ are assumed to be known, that is

$$t_i\ (T_s) = t_{oi} \qquad i = 1,2,\ldots, n \tag{E.5}$$

The problem of calculating the transient temperature distribution in a bearing arrangement thus becomes a problem of solving a system of non-linear differential equations of the first order with certain initial values given. The equations are non-linear since they contain terms of radiation and free convection, which are non-linear with temperature as will be shown later. The simplest and most economical way of solving these equations is to calculate the rate of temperature in crease at the time $T = T_k$ from equation E.4 and then calculate the temperatures at time $T_k + \Delta T$ from

$$t_{k+1} = t_k + \frac{dt_k}{dT}\Delta T = t_k + \frac{q_k}{\rho C_p V}\Delta T \tag{E.6}$$

If the time step ΔT used as program input is chosen too large, the temperatures will oscillate, and if it is chosen too small the calculation will be costly. It is therefore desirable to choose the largest possible time step that does not give an oscillating solution. The program optionally calculates such a time step. The step is obtained from the condition,

$$\frac{dt_{i,\ k+1}}{dt_{i,k}} \geq 0 \qquad i = 1,2,\ldots, n \tag{E.7}$$

If this derivation were negative, the implication would be that the local temperature at node i has a negative effect on its future value. This would be tantamount to asserting that the hotter a region is now, the colder it will be after an equal time interval. An oscillating solution would result.

Differentiating equation (E.6) for node i, one has as condition,

$$\frac{dt_{i,k+1}}{dt_{i,k}} = 1 + \frac{\Delta T_i}{\rho_i C_{pi} V_i} \cdot \frac{dq_{i,\,k}}{dt_{i,\,k}} \qquad i = 1, 2, \ldots n \qquad (E.8)$$

The derivative $dq_{i,k}/dt_{i,k}$ is calculated numerically

$$\frac{dq_{i,k}}{dt_{i,k}} = \frac{q_{i,k+1} - q_{i,k}}{\Delta t_i} \qquad (E.9)$$

For each node the value of ΔT_i giving a value of zero to the right hand side of Eqn. (E.8) is calculated.

A value of ΔT rounded off to one significant digit smaller than the smallest of the ΔT_i given by Eq. (E.8) is used.

If the transient thermal scheme is being used interactively with the bearing subprograms the user must specify a small enough time step between calls to the bearing subprograms in order that the variation in bearing generated heats, with time, accurately reflects the physical situation. At first a trial and error procedure will be required to effectively use the program in this mode, however, experience will increase the user's effectiveness.

E.4.3 Calculation of Heat Transfer Rate

The transfer of heat within a medium or between two media can occur by conduction, convection radiation and fluid flow.

All these types of heat transfer occur in a bearing application as the following examples show:

1. Heat is transferred by conduction between inner ring and post and between outer ring and housing.
2. Heat is transferred by convection between the surface of the housing and the surrounding air.

3. When the bearing is lubricated and cooled by circulating oil, heat is transferred by fluid flow.

Therefore, in calculating the net flow to a node all the above mentioned modes of heat transfer will be considered.

E4.3.1. Generated Heat

There may be a heat source at node i giving rise to a heat flow to be added to the heat flowing from the neighboring nodes.

In the case that the heat source is a bearing, it may either be considered to produce known amounts of power, in which case constant numbers are entered as input to the program, or the planet bearing program may be used to calculate the bearing generated heat as a function of bearing temperatures.

E.4.3.2. Conduction

The heat flow $q_{ci,j}$ which is transferred by conduction from node i to node j, is proportional to the difference in temperature $(t_i - t_j)$ and the cross-sectional area A and is inversely proportional to the distance ℓ between the two points, thus

$$q_{ci,j} = \frac{\lambda A}{\ell} (t_i - t_j) \qquad \text{(E.10)}$$

where λ = the thermal conductivity of the medium.

E. 4.3.3 Free Convection

Between a solid medium such as a metallic body and a liquid or gas, heat transfer is by free or forced convection. Heat transfer by free convection is caused by the setting in motion of the liquid or gas as a result of a change in density arising from a temperature differential in the medium. With free convection between a solid medium and air, the heat energy $q_{vi,j}$ transferred between nodes i and j can be calculated from the equation,

$$q_{vi,j} = \alpha_v A \, |t_i - t_j|^d \cdot \text{SIGN}(t_i - t_j) \qquad \text{(E.11)}$$

where α_v = the film coefficient of heat transfer by free convection

A = the surface area of contact between the media
d = is an exponent, usually = 1.25, but any value can be specified as input to the Program

$$SIGN = \begin{cases} 1 \text{ if } t_i \geq t_j \\ -1 \text{ if } t_i < t_j \end{cases}$$

The last factor is included to give the expression $q_{vi,j}$ a correct sign.

The value of α can be calculated for various cases, see Jacob and Hawkins, {16}.

E.4.3.4. Forced Convection

Heat transfer by forced convection takes place when liquid or gas moves around a solid body, for example, when the liquid is forced to flow by means of a pump or when the solid body is moved through the liquid or gas. The heat flow $q_{wi,j}$ transferred by forced convection can be obtained from the following equation:

$$q_{wi,j} = \alpha_w A(t_i - t_j) \qquad \text{(E.12)}$$

where α_w is the film coefficient of heat trasfer during forced convection. This value is dependent on the actual shape, the surface condition of the body, the difference in speed, as well as the properties of the liquid or gas.

In most cases, it is possible to calculate the coefficient of forced convection from a general relationship of the form,

$$N_u = aR_e{}^b P_r{}^c \qquad \text{(E.13)}$$

where a, b, and c are constants obtained from handbooks, such as {17} . R_e and P_r are dimensionless numbers defined by

Nu = Nusselt's number = $\alpha_W L/\lambda$
L = characteristic length
λ = conductivity of the fluid
R = Reynold's number = $UL\rho/n$
U = characteristic speed
ρ = density of the fluid
n = dynamic viscosity of the fluid
P_r = Prandtl's number = nC_p/λ
C_p = specific heat

The program can use a constant value of the coefficient of convection, or let it vary with actual temperatures, the variation being determined by how the viscosity varies. Input can then be given in one of three ways, for each coefficient.

Constant viscosity

1. Values of the parameters of equation (E.13) are given as input and a constant value of α_w is calculated by the program.

Temperature dependent viscosity

2. The coefficient α_w for turbulent flow and heating of petroleum oils is given by

$$\alpha_w = k_9 \cdot \eta(t)^{k_{10}} \qquad (E.14)$$

Where: k_9 and k_{10} are given as input together with viscosity at two different temperatures.

3. Values of the parameters of equation (E.13) are given as input. Viscosity is given at two different temperatures.

E.4.3.5 Radiation

If two flat parallel, similar surfaces are placed close together and have the same surface area A, the heat energy transferred by radiation between nodes i and j representing those bodies, will be,

$$q_{Ri,j} = \varepsilon\sigma A[(t_i + 273)^4 - (t_j + 273)^4] \qquad (E.15)$$

Where: ε is the surface emissivity. The value of the coefficient ε is an input variable and varies between 1 for a completely black surface and 0 for an absolutely clean surface. In addition σ is Stefan-Boltzmann's radiation constant which has the value 5.76×10^{-8} watts/$m^2 - (°K)^4$ and t_i and t_j are the temperatures (°C) at points i and j.

Heat transfer by radiation under other conditions can also be calculated, {16}. The following equation, for instance, applies between two concentric cylindrical surfaces

$$q_{Ri,j} = \epsilon\sigma \frac{A_i\,[(t_i + 273)^4 - (t_j + 273)^4]}{1 + (1-\epsilon)\,(A_i/A_e)} \quad (E.16)$$

Where A_i is the area of the inner cylindrical surface

A_e is the area of the outer cylindrical surface

E.4.3.6 Fluid Flow

Between nodes established in fluids, heat is transferred by transport of the fluid itself and the heat it contains.

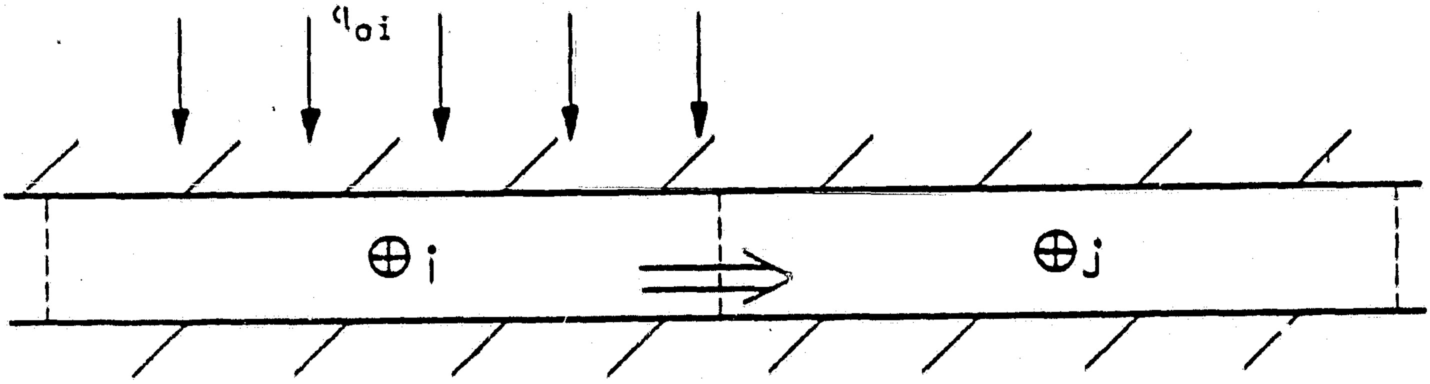


Figure E-5 Convective Heat Transfer

Figure E-5 shows nodes i and j at the midpoints of consecutive segments established in a stream of flowing fluid.

The heat flow $q_{ui,j}$ through the boundary between nodes i and j can be calculated as the sum of the heat flow q_{fi} through the middle of the element i, and half the heat flow q_{oi} transferred to node i by other means, such as convection.

The heat carried by mass flow is,

$$q_{fi} = \rho_i \; C_{p_i} \; V_i \; t_i = K_i \; t_i \quad (E.17)$$

Where V_i = the volume flow rate through node i

The heat input to node i is the sum of the heat generated at node i (if any) and the sum over all other nodes of the heat transferred to node i by conduction, radiation, free and forced convection.

$$q_{oi} = q_{G,i} + \sum_{j=1}^{m} (q_{ci,j} + q_{vi,j} + q_{wi,j} + q_{Ri,j}) \quad (E.18)$$

The heat flow between the nodes of Fig. E-5 is then,

$$q_{ui,j} = q_{fi} + q_{oi}/2 \quad (E.19)$$

If the flow from node i is dividing between nodes j and k, Figure E-6 then the heat flow is calculated from

$$q_{ui,j} = K_{ij} \, (q_{fi} + q_{oi}/2) \quad (E.20)$$

Where K_{ij} = the proportion of the flow at i going to node j, $0 < K_{ij} \leq 1$. K_{ij} is specified at input.

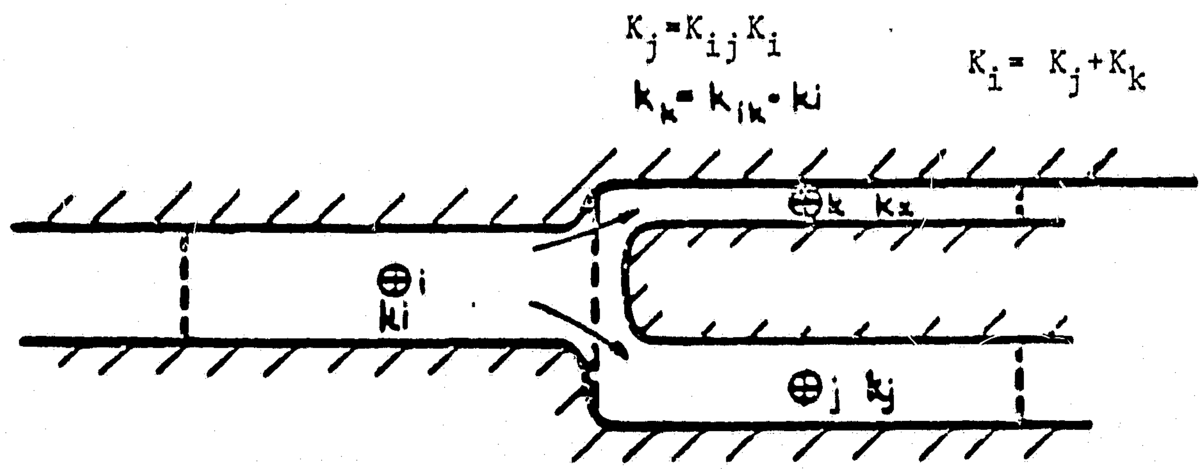


FIGURE E-6
DIVIDED FLUID FLOW FROM NODE i

E.4.3.7 Total Heat Transferred

The net heat flow rate to node i can be expressed as,

$$q_i = q_{G,i} + \sum_{j=1}^{m} \{q_{ci,j} + q_{ui,j} + q_{vi,j} + q_{wi,j} + q_{Ri,j}) \quad (E.21)$$

The summation should include all nodes j, both with unknown temperatures as well as boundary nodes, at which the temperature is known, so long as they have a direct heat exchange with node i.

This expression is a non-linear function of temperatures because of the terms q_W and q_R. Therefore the equations to be solved for a steady state solution are non-linear. The subprogram SOLVXX for solving non-linear simultaneous equations is used for this purpose.

E.4.4 Conduction Through a Bearing

As described in Section E.4.3.2 the conduction between two nodes is governed by the thermal conductivity parameter λ of the medium through which conduction takes place. The value of λ is specified at input.

An exception is when one of the nodes represents a bearing ring and the other a set of rolling elements. In this case the conduction is separately calculated using the principles described below.

E.4.4.1 Thermal Resistance

It is assumed that the rolling speeds of the rolling elements are so high that the bulk temperature of the rolling elements are the same at both the inner and outer races, except in a volume close to the surface. The resistance to heat flow can then be calculated as the sum of the resistance across the surface and the resistance of the material close to the surface.

The resistance is defined implicitly by

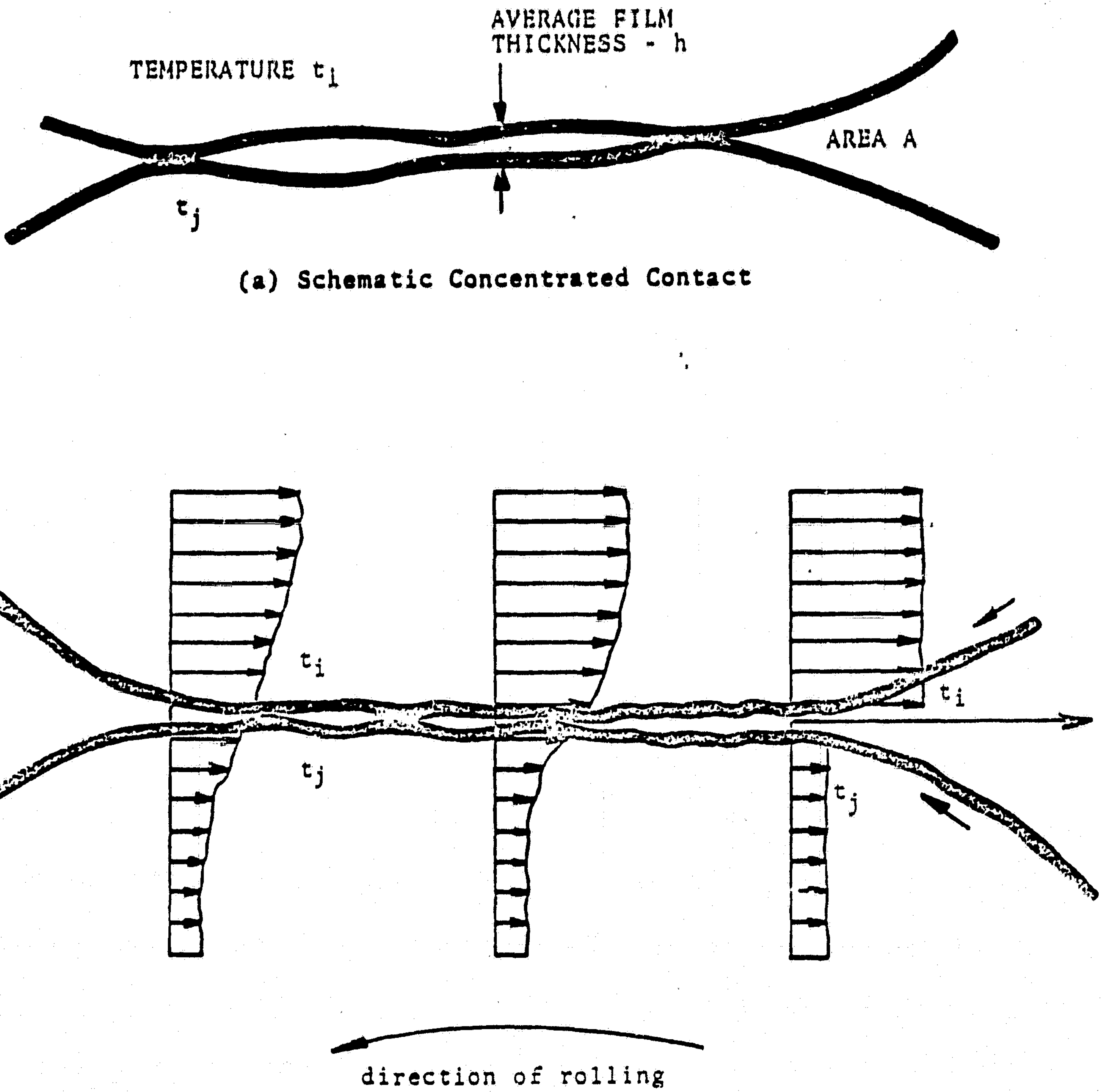
$$\Delta t = \Omega \cdot q \qquad (E.22)$$

where

Δt is temperature difference
q is heat flow

The resistance due to conduction through the EHD film is calculated as

$$\Omega_1 = h/(\lambda A) \qquad (E.23)$$



(b) Temperature Distribution at Rolling,
Concentrated Contact Surfaces

FIGURE E-7

CONTACT GEOMETRY AND TEMPERATURES

where h is taken to be the calculated plateau film thickness

A is the Hertzian contact area at the specific rolling element-ring contact under consideration

λ is the conductivity of the oil.

The geometry is shown in Figure E-7 (a).

So far, a constant temperature difference between the surface has been assumed. But during the time period of contact, the difference will decrease because of the finite thermal diffusivity of the material near the surface, Fig. E-7 (b).

To points at a distance from the surface this phenomenon will have the same effect as an additional resistance Ω_2 acting in series with Ω_1.

This resistance was estimated in {18}

$$\Omega_2 = \frac{1}{\lambda \ell_{re,i}} \left(\frac{\pi \psi}{2 b_i V} \right)^{1/2} \qquad (E.24)$$

where ℓ_{re} = contact length, or in the case of an elliptical contact area, 0.8 times the major axis

λ = heat conductivity

Ψ = thermal diffusivity $\lambda/(\rho C_p)$

ρ = density

C_p = specific heat

b = half the contact width

V = rolling speed

The resultant resistance is

$$\Omega_{res} = \Omega_1 + \Omega_2 \qquad (E.25)$$

There is one such resistance at each rolling element. They all act in parallel. The equivalent resistance,Ω eqv. is thus obtained from

$$\frac{1}{\Omega_{eqv}} = \sum_{i=1}^{n} \frac{1}{\Omega_{res,i}} \tag{E.26}$$

E.4.5 Consideration of Multiple, Axially Symmetric Subsystems within the Primary Thermal System.

The temperature mapping heat dissipation analysis is built on the assumption that the system is basically axially symmetrical so that each temperature node is essentially a circular ring with no temperature deviations around the circumference.

Solutions of the planetary system problem assumes that each of the planets within a stage transmits equal power and is subjected to identical load and speed conditions. This assumption permits the calculation of frictional heat for one planet bearing (one or two rows of rollers) but allows us to attribute the same frictional heat to each of the planets within the stage. The heat flow within a given planet is governed by its axially symmetric heat transfer characteristics.

The heat flow in the primary system is affected by the heat generated by all individual planets and its symmetry about another axis. To account for this the heat transfer coefficients between nodes in the prime and subsystem are direct and so that when heat flows from the primary system to the subsystem only (1/Npl) of the heat enters the subsystem where Npl is the number of planets. The inverse is true as heat leaves a subsystem and enters the primary system.

The subsystem-primary system heat transfer interaction is taken into account with program input, on card type T8.

APPENDIX F

"SKF" AND "NASA" VERSIONS OF FILM THICKNESS AND TRACTION FORCE CALCULATIONS

F-1. INTRODUCTION

This Appendix is a supplement to the User's Manual for SKF Computer Program PLANETSYS. It highlights the differences between the PLANETSYS/SKF and PLANETSYS/NASA program versions. In Section F-2 below, the differences between the mathematical models used in the two versions of the program are stated explicitly. These differences are related to:

1) The calculation of EHD film thickness
2) Concentrated contact traction

In section F-3, the differences between the two versions in terms of program execution are specified.

F-2. MATHEMATICAL MODELS

F-2.1 EHD Film Thickness

PLANETSYS/SKF uses the Dowson Higginson (13) equation to calculate film thickness for line contact. The calculated value is multiplicatively modified by a film thickness thermal reduction factor developed by Cheng (8) and by a starvation reduction factor developed by Chiu (9).

PLANETSYS/NASA uses the equation developed by Loewenthal et al (21) in calculating film thickness, for both point and line contact.

F-2.2 Concentrated Contract Traction

The concentrated contact traction model used in PLANETSYS/SKF accounts for lubricant shear and asperity interaction. A semi-empirical model (10), is used to calculate an EHD lubricant shear coefficient. Asperity effects are introduced by determining the portion of the contact load carried by the asperities, using the analysis of Tallian (11), and then calculating the resulting traction as the product of the normal load carried by the asperities times the asperity friction coefficient. In equation form the traction force is:

$$F = Q_{EHD}\,\mu_{EHD} + Q_{ASP}\,\mu_{ASP} \qquad \text{(F-1)}$$

$$Q = Q_{EHD} + Q_{ASP} \qquad \text{(F-2)}$$

Q is the normal load

Q_{EHD} is the normal load carried by the EHD film

μ_{EHD} is the friction coefficient which develops from lubricant shear

Q_{ASP} is the normal load carried by asperities

μ_{ASP} is the asperity friction coefficient

F is the traction force

PLANETSYS/NASA calculates concentrated contact traction across the EHD film only, according to the model developed by Allen, et al (22). This model calculates the traction force by first calculating the shear stress according the the Newtonian fluid shear equation.

$$\tau = \eta \frac{v}{h} \qquad \text{(F-3)}$$

where τ is the shear stress
η is the dynamic viscosity
v is the surface relative sliding velocity
h is the lubricant film thickness

The lubricant viscosity is assumed to be an exponential function of pressure of the form:

$$\eta = \eta_o e^{\alpha P} \qquad \text{(F-4)}$$

where η_o is the dynamic viscosity at atmospheric pressure

α is the pressure viscosity coefficient defined implicitly above

P is the normal pressure

Allen requires that the shear stress not exceed a specified fraction of the normal stress such that

$$\tau = \eta \frac{V}{\bar{h}} \quad \text{if } \eta \frac{V}{\bar{h}} \leq \tau_c \tag{F-5}$$

$$\tau = \bar{f}\bar{P} \text{ if } \eta \frac{V}{\bar{h}} > \bar{f}\bar{P} \text{ and } \eta \frac{V}{\bar{h}} > \tau_c \tag{F-6}$$

where, τ_c is the critical shear stress for which a value of 0.0069 N/mm^2 (1000 psi) is used.

$\bar{f}$ is called the lubricant friction coefficient and has been determined for specific lubricants. Typical values of $\bar{f}$ lie in the range $0.05 \leq \bar{f} \leq 0.08$.

Having calculated the shear stress, the traction force is obtained by integrating the shear stress over the respective contact area.

F-3. SKF VS. NASA VERSIONS - PROGRAM USE

The selection of the desired PLANETSYS version has been made possible by the inclusion of two separate Map Statements for the Univac 1100 computer:

1) The original map statement
@MAP,S R,A
enables an execution with the SKF film thickness and traction models. Subroutines FMIXPL, EHDSKF, and FRICTN are called to compute the EHD traction forces.

2) A new map statement
@MAP,S RNASA,ANASA
enables the use of the NASA film thickness and traction models. Subroutines FMIXPL/NASA, FILM, and ALLEN are called to compute the EHD traction forces.